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HYDRAULIC NOISE STUDY

Oklahoma State University

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Prepared for:

Army Mobility Equipment Research and Development Center

December 1974

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HYDRAULIC SYSTEM NOISE STUDY

Prepared By Personnel of

FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

December 1974

ANNUAL REPORT

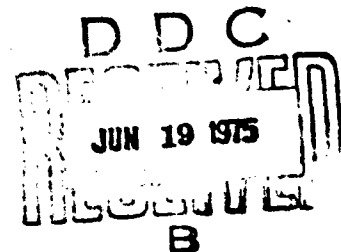
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number) The purpose of the Oklahoma State University/U.S. Army Mobility Equipment Research and Development Center Program is to provide the military with tools for the scientific appraisal of fluid power systems. The activities of the fourth year of the noise program are a continuation of the efforts of the previous years and make full use of the preceding program.		

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This is the second section of the complete annual report, which is divided into the following five self-contained sections:

- FPRC-4M1 Hydraulic System Controls Study
- FPRC-4M2 Hydraulic Noise Study
- FPRC-4M3 Lubricating Oil Filtration Study for Mobile On-Off Highway Diesel Engine Driven Vehicles
- FPRC-4M4 Contamination Control
- FPRC-4M5 On-Board Hydraulic System Monitor Study

This report presents the results of a hydraulic noise study which experimentally examined the sound power of selected military components and investigated the effectiveness of noise control techniques for fluid power systems.

The component sound power measurements are compared to similar measurements for other components. A pump noise model is proposed which could minimize component acoustical testing.

Techniques for controlling hydraulic system pressure ripple, reservoir noise, structure-borne noise from pumps, and conduit noise are discussed in the report. It is shown that pressure ripple attenuators, vibration isolators, and damping compound offer three techniques for controlling fluid power system noise.

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— FOREWORD —

This report was prepared by the staff of the Fluid Power Research Center of the School of Mechanical & Aerospace Engineering, Oklahoma State University of Agriculture and Applied Sciences. The study was initiated by the Mobility Equipment Research and Development Center, Fort Belvoir, Virginia. Authorization for the study reported herein was granted under Contract No. DAAK02-72-C-0172. The time period covered by this report is 9 November 1973 to 8 November 1974.

The Contracting Officer's Representative was Mr. Hansel Y. Smith, and Mr. John M. Karhnak served as the Contracting Officer's Technical Representative. In addition, Mr. Paul Hopler has effectively represented the Contracting Officer both technically and administratively through various phases of this contract. The active participation of Messrs. Smith, Karhnak, and Hopler during critical phases of work contributed significantly to the overall success of the program. Project members are grateful for the assistance and guidance of Mr. S.E. Wehr, U.S. Army MERDC.

This report is one of the five self-contained sections into which the annual report has been divided. The titles of the various sections are listed below:

- SECTION I: HYDRAULIC SYSTEMS CONTROL STUDY
- SECTION II: HYDRAULIC NOISE STUDY ✓
- SECTION III: LUBRICATING OIL FILTRATION STUDY FOR MOBILE ON-OFF HIGHWAY
DIESEL ENGINE DRIVEN VEHICLES
- SECTION IV: CONTAMINATION CONTROL
- SECTION V: ON-BOARD HYDRAULIC SYSTEM MONITOR STUDY

The study represented by this report was conducted under the general guidance of Dr. E. C. Fitch, Program Director. Mr. G. E. Maroney served as the Program Manager for

the area. Messrs. L. R. Elliott and T. G. Snyder performed most of the experimental work and were assisted by Messrs. S. E. Smith, J. C. Boydston, and D. L. O'Neal.

This report presents the results of a hydraulic noise study which experimentally examined the sound power of selected military components and investigated the effectiveness of noise control techniques for fluid power systems.

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TABLE OF CONTENTS

<i>Chapter</i>		<i>Page</i>
I	INTRODUCTION	1
II	SYSTEM NOISE	3
III	NOISE CONTROL	7
IV	PUMP NOISE	11
	Pump Noise Model	11
	Pump Noise Rating	13
	Pump Sound Vs. % Air	16
	Pump ABN Isolation	18
	Pump SBN Isolation	19
V	CONDUIT NOISE	21
VI	RESERVOIR NOISE	29
	Structureborne Noise	29
	Fluidborne Noise	32
VII	FLUIDBORNE NOISE CONTROL	35
	PULSCO	35
	Pulse-Tone	38
VIII	SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS	43
 <i>Appendices</i>		
A	TEST PROCEDURES	47
B	ACOUSTICAL DATA REDUCTION	53
C	SELECTED REFERENCES	59
D	INSTRUMENTATION	63
E	MATERIALS AND COMPONENTS	69

LIST OF TABLES

<i>Table</i>		<i>Page</i>
2-1	Tabular Model for Estimating and Controlling Fluid Power System Noise ..	5
3-1	Noise Generation, Transmission, Emission, and Control Relationships in Fluid Power Systems	10
4-1	Sound Power Levels of Untreated Pumps (re 10^{-12} watts) and Fluidborne Noise Levels (re $20\mu\text{N}/\text{M}^2$)	12
4-2	Pump Sound Power Data with Model Parameters	14
4-3	Test Results Using Damping Materials on Pumps to Reduce ABN. Sound Power dbA.	19
4-4	Summary of SBN Isolation Tests	20
5-1	Airborne Conduit Sound Power db/dbA as a Function of ABN Isolator Treatment	26
6-1	Summary of Structureborne Noise Control Study on 6000 lb RTFL Reservoir. Sound Power Level (db)	30
6-2	Summary of Sound Power Data for Small Reservoir. Input Displacement Constant	31
7-1	Overall Changes in dBA Levels with PULSCO Installed	36

LIST OF FIGURES

<i>Figure</i>		
2-1	Example Fluid Power System with Component Labels for Noise Identification	4
3-1	Basic Progression of Noise Showing Relationship Between Fluidborne Noise, Structureborne Noise, and Airborne Noise	7
3-2	Basic Acoustical Interactions Between Components Which Affect Noise Generation	8
3-3	Noise Control Processes of Absorption and Reaction	9
4-1	Pump Noise Chart	15
4-2	Component Noise Evaluation Summary	16
4-3	Frequency Spectrum for "Normal" Operating Conditions	17
4-4	Frequency Spectrum for Pump Operating with Inlet Aeration Due to Low Inlet Pressure	17
4-5	Frequency Spectrum for Pump Operating with Inlet Aeration Due to Injected Air	17
4-6	Cross-Section of Pump Isolation Technique	20

LIST OF FIGURES

<i>Figure</i>		<i>Page</i>
5-1	Illustration of Two Sources of Conduit Noise – Vibration and Pressure Ripple	21
5-2	Sound Power Vs. Frequency for Conduit with PULSCO Installed Downstream of Conduit. Pump Speed 1500 rpm, Outlet Pressure 2000 psi	22
5-3	Sound Power Vs. Frequency for Conduit with PULSCO Installed Downstream of Conduit. Pump Speed 2000 rpm, Pressure 2000 psi	23
5-4	Sound Power Vs. Frequency for Conduit without PULSCO Installed. Pump Speed 1000 rpm, Pressure 2000 psi	24
5-5	Sound Power Vs. Frequency for Conduit without PULSCO Installed. Pump Speed 1500 rpm, Pressure 2000 psi	25
5-6	Sound Power Vs. Frequency for Conduit without PULSCO Installed. Pump Speed 2000 rpm, Pressure 2000 psi	27
6-1	Vibration Test Fixture	30
6-2	Effects of Reservoir Noise Reduction Techniques Applied to Small Reservoir. Input Displacement Constant dbA Levels Are Summed Between 100 Hz and 1000 Hz	32
6-3	Sound Level Changes of Filter Test Stand Due to Modifications	34
7-1	Fluidborne Noise Levels in Conduit for Various Test Conditions with PULSCG. Pump NP-1, 2000 rpm, 2000 psi	37
7-2	Conduit Sound Power for Various Test Conditions with PULSCO. Pump NP-1, 2000 rpm, 2000 psi	38
7-3	Comparison of the FBN Transmission Loss Obtained with Two Types of FBN Attenuators	39
7-4	Data for 2000 rpm Pump Speed, 15 in ³ Pulse-Tone, 2000 psi System Pressure, 1000 psi Precharge	39
7-5	Pulse-Tone Fluidborne Noise Controller Evaluation Results [7].	40
7-6	PULSCO FBN Controller Published Characteristics and Test Results [9].	40
A-1	Comparison of FBN Levels Near Pump and Approximately 1 ft. from Pump (db re 20μ/m ²)	50
A-2	FBN Level Changes Due to Load Valve Location	51

CHAPTER I

INTRODUCTION

The objectives of this noise study were to measure the acoustical characteristics of selected hydraulic components from the 6000 lb. rough terrain fork lift and to examine practical fluid power noise reduction techniques for mobile equipment. The ultimate objective of these efforts is the control of noise related to fluid power systems.

Noise is frequently defined as "*any undesired sound.*" By definition then, there must be a sound receiver who judges the sound to be noise, a transmission path for the sound to reach the receiver, and a sound source. Controlling the noise involves modifying the source or the transmission path. Although it is common, in extreme cases, to isolate the receiver, this procedure was not addressed in this report because most legislation places limits on a maximum sound level without regard to possible receiver protective devices.

Chapter II introduces a basic noise model for a hydraulic system. This simple model provides a tabular approach to account for the multiplicity of sound sources and sound transmission paths associated with hydraulic systems. Chapter III examines the basic approaches available to control the noise associated with fluid power systems. The remainder of the report presents and discusses the results of using various noise control techniques with selected fluid power components and systems.

Fluid power pump noise and its control is the subject of Chapter IV. The chapter discusses a model for estimating the sound power of pumps. The estimates are based on the model and at least three experimental data points. The influence of selecting a quiet noise source is emphasized. The chapter also discusses the effects of entrained air on pump sounds, pump airborne noise isolation, and pump structureborne noise isolation.

Chapters V and VI are concerned with the control of conduit and reservoir noise, respectively. Conduit noise control is examined through the use of fluidborne noise reduction and the isolation of conduit airborne noise. Structureborne noise isolators and damping compounds are examined as a means of reducing the noise associated with hydraulic reservoirs.

Two devices for attenuating system fluidborne noise are discussed in Chapter VII. Test results are presented which show the differences between the two types of attenuators. Performance parameters of pressure ripple attenuation versus frequency and pressure drop versus flow are recommended as a result of the studies.

Chapter VIII summarizes the more pertinent results of the study, makes specific recommendations for noise control in hydraulic systems, and suggests areas for development that have the most immediate potential for reducing fluid power system noise levels.

The appendices contain information about the test procedures used to obtain experimental data, selected references, instrumentation used during the study, components evaluated, and materials used for absorption and isolation.

This report contains many summaries of the data which convey the test results. Generally, these results are given in dBA or db at specific frequencies. It was considered impractical to convey all of the individual test results at 1/3 octave frequencies because the document would have become unwieldy. Some of the more pertinent test results (such as pump sound power level data) have been reproduced and forwarded to the appropriate U.S. Army MERDC representatives under separate cover.

CHAPTER II

SYSTEM NOISE MODEL

In order to effectively address any control problem, it is necessary to formulate a reasonably accurate description of the control problem. In the absence of explicit system models (which are seldom available for complex practical problems), it is necessary to establish the basic factors which cause the system output and, further, to evaluate the interactions between the system variables in order to affect the desired control. Hopefully, repeated application of this process will lead to explicit relationships which nudge the art of engineering slightly closer to the science of engineering.

Noise control in a fluid power system requires examining the various components which transmit energy to other components or to the air. Fig. 2-1 illustrates a simple fluid power system whose components are numbered for identification. Each component in the example system either transmits energy to other components or to the air. Thus, it is necessary to account for the airborne noise (abn) produced by each component. Airborne noise is the result of vibration or structureborne noise (sbn). It is necessary to account for the sbn of each component, since this energy either becomes abn or is transmitted to other components as sbn. Finally, since one source of sbn is the pressure ripple in fluid power systems, it is necessary to account for the fluidborne noise (fbn) produced or transmitted by components. The "frame" or machine structure represents an important component because it transmits sbn between hydraulic components and from other machine systems to the hydraulic components.

Table 2-1 represents a tabular technique for modeling the noise associated with fluid systems. The tabular model may be used to provide a qualitative assessment of the hydraulic system noise. If all of the implied interactions are known, a realistic assessment of the sound power emitted by the system can be obtained. The use of Table 2-1 requires an appraisal of the circuit (such as shown in Fig. 2-1) combined with the known characteristics of the system components.

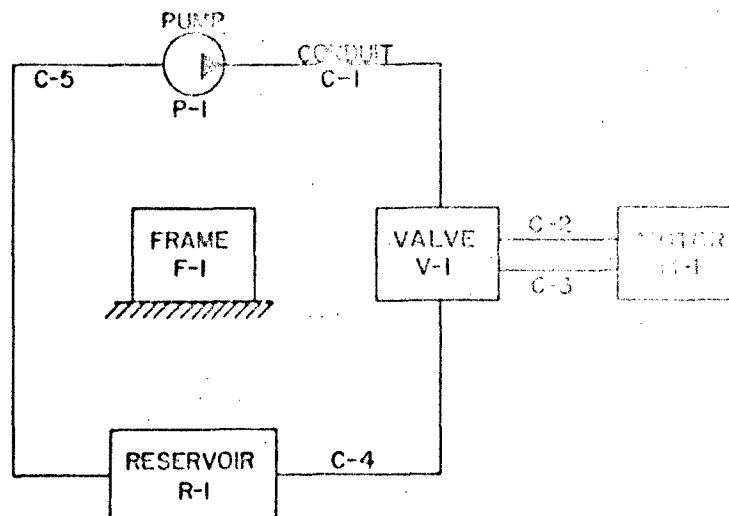


Fig. 2-1. Example Fluid Power System with Component Labels for Noise Identification.

Noise from any component is due to direct radiation or secondary effects (interactions with energy from another source). All of the components can be categorized primarily as either active or passive elements. An active element is one whose primary function is to convert energy from one form to another (the pump converts mechanical energy to hydraulic energy). A passive element primarily transmits energy between components (the conduit transmits fluid power between components) or serves primarily to condition the fluid (the reservoir may serve as a heat exchanger and air release device). Generally, the active elements will be the primary generators of noise, and the passive elements radiate noise due to interactions.

The first component listed in Table 2-1 is the pump (p-1). Because the pump is converting energy, it is basically an active device. Although it will undoubtedly emit some noise due to interactions with the frame or other components, the most important consideration for noise control is the noise directly generated by the pump. Therefore, for p-1, the tabular model ignores any noise that is due to interactions and concentrates on the directly emitted abn and the generated fbn and sbn. The abn noise directly emitted by the pump is entered in column six of the table. To obtain an effective sound power

TABLE 2-1. TABLE MODEL FOR ESTIMATING AND CONTROLLING FLUID POWER SYSTEM NOISE

HYDRAULIC COMPONENT	DIR	FLOW (GPM)	NOISE TYPE			NOISE ATTENUATION FACTOR	SOUND POWER ABN
			FLOW	SEN	ABN		
PUMP P-1	✓	✓	212	(7)	90	-4	86
CONDUIT C-1	✗	F-1	212	(7)	82	0	82
	✗	V-1	200	(7)	70	0	70
	✗	F-1	✗	(neg)	(neg)	—	—
CONDUIT C-2	✗	V-1	260	(7)	70	0	70
	✗	M-1	200	...	75	0	75
	✗	F-1	✗	(neg)	(neg)	—	—
VALVE V-1	✓	✗	200	(7)	80	-3	77
MONITOR M-1	✓	✗	200	(7)	94	-2	92
RESERVOIR R-1	✗	F-1	✗	(7)	85	0	85
	✗	C-4	180	(7)	50	0	50
	✗	C-5	200	(7)	70	0	70
FRAME F-1	✗	F-1	✗	(7)	(neg)	—	—
	✗	C-1	✗	(7)	(neg)	—	—
	✗	R-1	✗	(7)	(neg)	—	—

(neg) — negligible

(7) — not available

5

TOTAL SOUND POWER

94

level for the pump relative to the total machine level, there may be an attenuation factor. The attenuation may be due to acoustical isolation or other acoustical treatment. In the example, an attenuation of 4 db was used to obtain the final value in Column 8.

Each of the system components must be considered in the same manner as P-1. For instance, the conduit C-1 could produce noise due to several interactions as shown in the table. In order to assess the abn from C-1 due to P-1, a value for the fbn produced by P-1 is entered in the table. If it is known, a value of sbn induced in C-1 due to P-1 is also entered in the table. The interactions between the fbn and sbn from P-1 to C-1 can then be evaluated to yield an estimate of the abn from C-1 due to P-1. The figures in Table 2-1 are hypothetical values based on the results of this year's study and previous MERDC-OSU noise studies [1][2][3].

Once values for the tabular sound model have been estimated, it is possible to estimate the total contribution of fluid power systems to the machine noise level by summing the individual sound power estimates as illustrated in Table 2-1.

The availability of a model of the system noise allows the designer to establish the need for noise control and, if it is needed, rationally select the more significant sources of noise which should be attacked initially in the noise control effort.

CHAPTER III

NOISE CONTROL

If it becomes necessary to reduce the noise level of the hydraulic system without changing the operating conditions, there are two options available:

1. Change components (includes modifying the present one).
2. Add noise control components.

Fig 3-1 illustrates the basic relationship between fbn, sbn, and abn. If the magnitude of the original noise source (such as pressure ripple) cannot be reduced, then something must be added to the system to attenuate the noise-producing energy before it reaches the receiver.

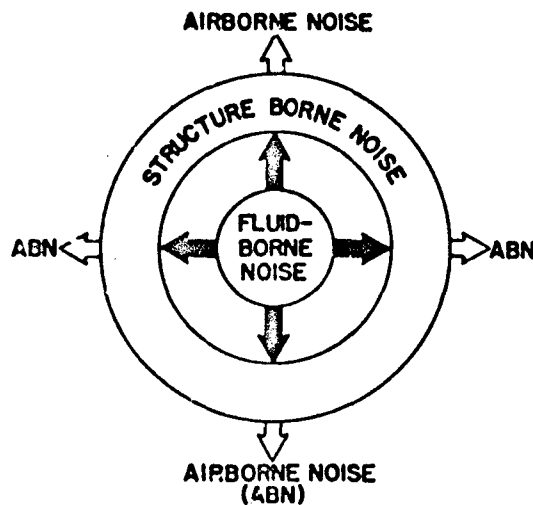


Fig. 3-1. Basic Progression of Noise Showing Relationship Between Fluidborne Noise, Structureborne Noise, and Airborne Noise.

Fig. 3-2 illustrates some of the interactions that must be considered when attempting to reduce fluid power noise. Reference to the tabular model in Chapter II gives a more comprehensive picture of the best areas to attack to achieve fluid power system noise control. In most systems, it is highly probable that the pump is an excellent candidate for noise reduction. But, it is also possible that the pump being used is one of the quietest available, and it might be impractical (for

financial or other reasons) to seek a "quieter" hydraulic power supply. In the latter case, if the pump is producing excessive fbn or sbn, then the noise can be reduced, usually by adding other components to the system. The additional components would be noise control elements.

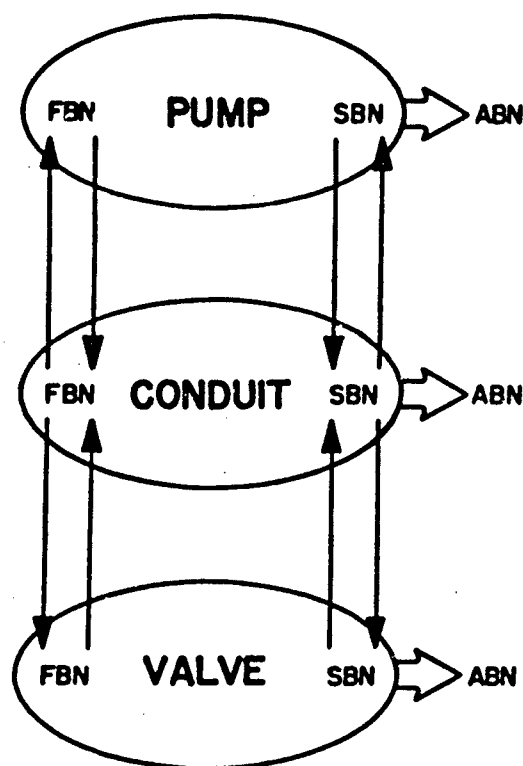


Fig. 3-2. Basic Acoustical Interactions Between Components Which Affect Noise Generation.

Fig. 3-3 illustrates the two basic noise control processes available – absorption and reaction. The two processes generally occur together. In other words, no noise reduction process is totally absorptive or totally reactive. However, in many cases, at certain frequencies, a particular process may be viewed as being primarily absorptive or primarily reactive. The reduction of pressure ripple by causing the fluid flow to go through a porous media is primarily an absorptive process, illustrated in Fig. 3-3(a). An accumulator is primarily a reactive device, since it relies on a phase relationship between the incident wave and the branch wave to minimize the acoustic wave traveling downstream (see Fig. 3-3(b)).

It is important to note that, for both processes illustrated in Fig. 3-3, it is implied that a portion of the energy incident at the controller reflects back upstream. This reflected energy frequently combines with the incident wave to cause standing wave patterns upstream of the noise controller. Because of the possibility of standing waves occurring between the real noise source and the noise controller, many manufacturers of noise control devices recommend mounting the noise controller directly to the source.

Table 3-1 outlines some of the sources of noise in a fluid power system and the types of noise controllers that might be used to control any undesired sound emitted by those sources. This study examines fluidborne, structureborne, and airborne noise controllers that incorporate both absorptive and reactive characteristics. The following chapters vividly illustrate that the proper selection of quiet components and the judicious use of noise controllers can significantly reduce fluid power system noise.

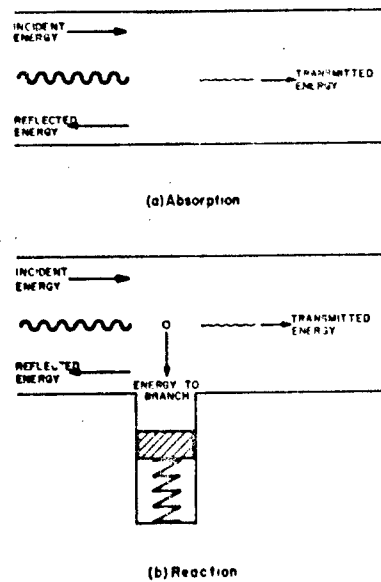


Fig. 3-3. Noise Control Processes of Absorption and Reaction.

TABLE 2-1. NOISE GENERATION, TRANSMISSION, EMISSION, AND CONTROL RELATIONSHIPS IN FLUID POWER SYSTEMS.

	HYDRAULIC COMPONENT	NOISE CATEGORY		
		FLUIDBORNE	STRUCTUREBORNE	AIRBORNE
NOISE SOURCES	PUMP	GENERATION	GENERATION TRANSMISSION	EMISSION
	CONDUIT	TRANSMISSION	TRANSMISSION	EMISSION
	VALVE	GENERATION TRANSMISSION	GENERATION TRANSMISSION	EMISSION
	ACTUATOR	GENERATION	GENERATION TRANSMISSION	EMISSION
NOISE CONTROLLERS	FLUIDBORNE NOISE CONTROLLERS	ABSORPTION REACTION	TRANSMISSION	-----
	STRUCTUREBORNE NOISE CONTROLLERS	-----	ABSORPTION REACTION	-----
	AIRBORNE NOISE CONTROLLERS	-----	TRANSMISSION	ABSORPTION REACTION

CHAPTER IV

PUMP NOISE

Hydraulic pumps are one of the major sources of acoustic energy in fluid power systems. Even if they do not directly radiate a significant quantity of sound power, the pressure ripple and structural vibrations they generate can indirectly cause a significant amount of airborne noise in a machine system. This chapter discusses the general character of pump noise, a model for pump noise, technique for apprising the sound power of hydraulic pumps, the effects of entrained air on pump sound level, the results of tests using damping compound on pumps, test results using airborne noise isolators, and the results of tests using vibration isolators on pumps.

PUMP NOISE MODEL

It has been stressed that the noise generation characteristics of a pump should be evaluated. Table 4-1 shows the results of sound power and fluidborne noise measurements on five different components used on the 6000 lb. rough terrain fork lift. These data, acquired as part of this study, show that components 36 and 37 emit more abn and generate more fbn than components 30, 31, and 32. In order to effectively assess the noise characteristics of these components, the data given in Table 4-1 must be compared with noise data from similar pumps. After making a comparison, it can be determined if the pump can be considered a "quiet" source.

There are several ways to assess the airborne noise characteristics of a pump:

1. Compare noise test results of two pumps on a point-per-point basis.
2. Graph the results of noise tests and compare sound power graphs of different units.

TABLE 4-1. SOUND POWER LEVELS OF UNTREATED PUMPS (re 10^{-12} watts) AND FLUIDBORNE NOISE LEVELS (re $20 \mu\text{N}/\text{M}^2$).

OSU-NP NO.	SPEED (rpm)	PRESSURE (psi)	HP	ABN (dba)	FBN (dba)
30	2000	2000	7.2	75.7	206.1
	2000	200	0.8	66.1	194.7
	600	2000	2.3	65.8	195.0
	600	200	0.3	53.1	179.3
31	2000	2000	6.8	74.7	205.5
	2000	200	0.7	64.6	196.0
	600	2000	3.9	63.8	196.4
	600	200	0.3	53.7	177.8
32	2000	2000	7.5	74.0	205.0
	2000	250	1.1	66.4	198.4
	600	2000	2.3	62.9	193.6
36	2000	2000	14.6	96.7	221.6
	2000	200	2.8	81.6	207.4
	600	1800	3.9	79.6	209.9
	600	200	1.2	63.0	188.6
37	2000	2000	16.6	94.1	226.4
	2000	200	2.7	78.8	207.9
	600	2000	5.4	75.7	208.3
	600	200	1.1	66.0	189.6

OSU-NP 30, 31, & 32 Ref. V1001S4 S 11D20L

OSU-NP 36 & 37 Ref. V2F1F9S18B8H20A001L

3. Make comparisons of pumps using a pump noise rating method.
4. Evaluate pump noise characteristics using coefficients from a pump noise model.

Each of these assessment techniques can be better understood if a model for the noise emission characteristics of a pump is studied.

A pump sound power level model was developed for this study. The model proposes that a pump's sound power can be described using the following equation [4]:

$$W_A = K_1 N^\alpha 10^{\beta P} \quad (4-1)$$

where: W_A = sound power level in watts ("A" weighted)
 K_1 = pump characteristic
 N = speed, rpm
 α = pump characteristic
 β = pump characteristic
 P = pump outlet pressure, psi

Since α generally has a value of about 2.0 and β has an order of magnitude of 10^{-4} , it can be shown that the pump sound power increases more rapidly as a function of speed than it increases as a function of pressure. The most important point to be made here is that the model smoothes the actual experimental data, which may hide some of the peak values of the pump sound that do not follow the assumed model. This characteristic of the model – the fact that it smoothes the test data – need not cause great alarm, but it should be recognized. One of the most attractive reasons for using a model for pump noise is that the model allows estimating the pump's sound power over a wide operating range based on a minimum number of tests which are used to identify the pump characteristics K_1 , α , and β .

The pump noise model is discussed more fully in Ref. [4]. It is shown that, for two pumps considered in the paper, the pump sound power model estimates the actual sound power of the pumps within 0.7 dBA.

PUMP NOISE RATING

If it is desired to compare pumps on a point-to-point basis, the comparison can be accomplished by referring to a table of data such as Table 4-2. Another approach for comparing similar pumps at the same pressure and speed would be to make graphs of the pump's sound power levels, such as Fig. 4-1, and make evaluations between graphs. The graphs could be made from numerous data points or could be made using the pump noise model with as few as three data points.

TABLE 4-2. PUMP SOUND POWER DATA WITH MODEL PARAMETERS.

OSU-NP TYPE	SPEED (rpm)	PRESSURE (psi)	HYDRAULIC POWER (watts)	ABN POWER (dba)	AEN (watt, A)	PNR 10 ⁹	β 10 ⁴	K
29 Screw	1000	1000	1262.0	65.8	3.80E-6	3.01	1.6	
	2000	200	1436.0	70.2	1.05E-5	7.33	0.2	
	2000	2000	7138.0	70.6	1.15E-5	1.62		6.2E-11
30 Vane A	600	200	217.6	53.1	2.04E-7	0.938	2.1	
	600	2000	1741.0	65.8	3.77E-6	2.160	5.8	
	2000	200	565.8	68.1	4.07E-6	7.200		
	2000	2000	5397.0	75.7	3.72E-5	6.880		2.8E-12
	2000	2000	5397.0	72.32	1.71E-5	3.170		
31 Vane	600	200	217.6	53.7	2.35E-7	1.08	2.1	
	600	2000	2873.0	63.8	2.37E-6	0.825		
	2000	200	548.4	64.5	2.85E-6	5.20	5.3	
	2000	2000	5049.0	74.7	2.92E-5	5.79		3.0E-13
32 Vane	600	2000	1741.0	62.9	1.93E-6	1.11	2.1	
	2000	250	848.7	66.4	4.33E-6	5.11	4.7	
	2000	2000	5571.0	74.0	2.49E-5	4.47		2.7E-13
33 External Gear A R	1200	2500	5658.0	74.3	2.69E-5	4.75	3.1	
	1800	250	968.4	78.4	6.95E-5	71.80		
	1800	2500	9140.0	77.8	6.01E-5	6.58	-0.3	
	2400	2500	12187.0	82.3	1.70E-4	13.90		
	1200	2500	5658.0	67.6	5.75E-6	1.02		
	1800	2500	9249.0	78.0	6.34E-5	6.65		4.8E-15
36 Vane	600	200	870.5	63.0	2.01E-6	2.31	3.2	
	600	1800	2977.0	79.6	9.16E-5	30.80		
	2000	200	2063.0	81.6	1.46E-4	70.70	9.4	
	2000	2000	10881.0	96.7	4.65E-3	430.00		1.3E-15
37 Vane	600	200	827.0	66.0	3.98E-6	4.81	3.5	
	600	2000	4004.0	75.7	3.74E-5	9.34		
	2000	200	1984.7	78.8	7.55E-5	38.00	6.9	
	2000	2000	12361.0	94.1	2.56E-3	207.50		5.5E-16
40 External Gear R A	1000	200	748.0	71.3	1.35E-5	18.00	2.2	
	2000	100	744.0	78.1	6.45E-5	86.70	-3.9	
	2000	200	1506.0	77.7	5.89E-5	30.10		
	2000	200	1506.0	77.9	6.16E-5	40.90		5.6E-12
	2000	200	1506.0	72.9	1.98E-5	12.90		

R = Repeatability

A = Acoustical Isolator

Another approach for evaluating the sound performance of hydraulic pumps is to use a rating number, PNR (Pump Noise Rating). The PNR is the ratio of the sound power produced by a pump to the hydraulic power produced by the pump. Expressed mathematically, this is:

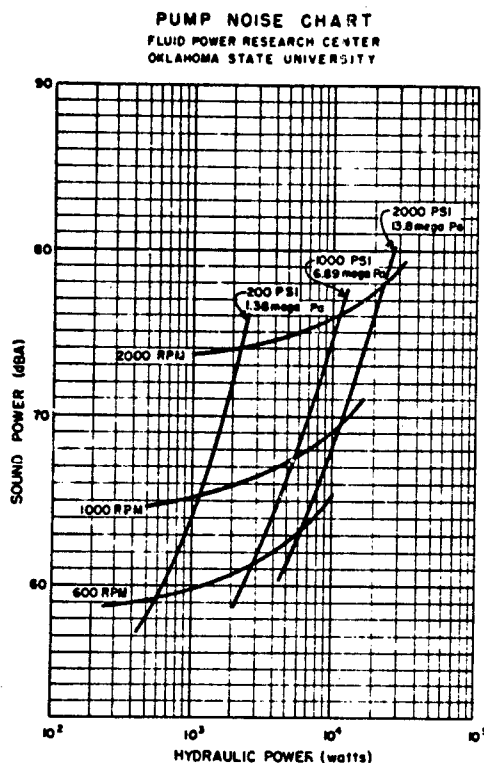


Fig. 4-1. Pump Noise Chart.

$$PNR = W_A / W_H = \frac{K_1 N^{\alpha} 10^{\beta P}}{K_2 P Q} \quad (4-2)$$

where:

W_H = hydraulic power, watts

K_2 = constant

Q = flow rate, litres/second

Fig. 4-2 is a graph of PNR versus hydraulic power for several pumps. Note that Fig. 4-2 is limited to a fixed set of test conditions. The reason for plotting PNR at one set of test conditions is the fact that PNR, like pump sound power in general, is nonlinear. If it is assumed that Q is a linear function of N , the equation (4-2) can be rewritten:

$$PNR = K_3 N^{\alpha-1} 10^{\beta P} / P \quad (4-3)$$

where K_3 is a new constant. This indicates that, at a constant pressure, the PNR increases with increasing speed. At a constant speed, the PNR will have some minimum point as a function of pressure. In other words, at a given speed, each pump has some particular pressure at which it generates the least amount of sound relative to the total horsepower that is being delivered.

The coefficient K_1 in Eq. (4-1) offers an exciting possibility of being an excellent rating number for pump sound power. Although K_1 will not allow discrimination between pumps when its order of magnitude is the same for different pumps, in general, the range of K_1 is great enough that it will allow discrimination between different pumps. Certainly,

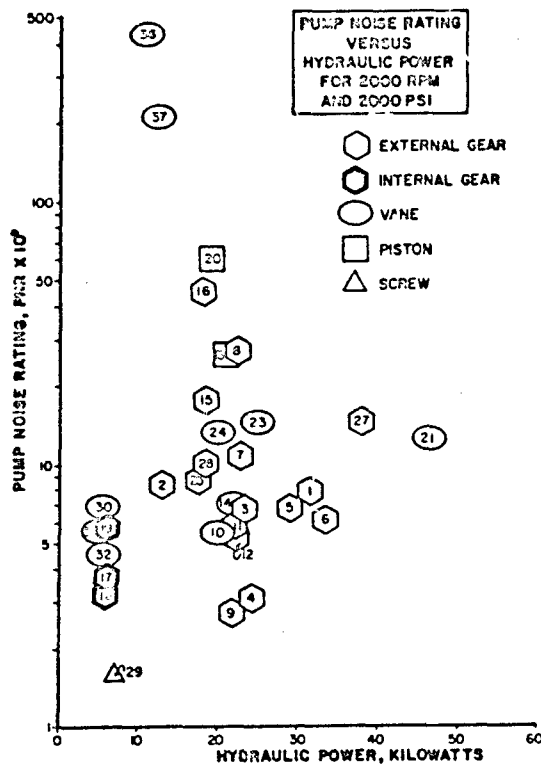


Fig. 4-2. Component Noise Evaluation Summary.

no entrained air is visible at the inlet to the pump [5]. The "all-pass" level, total pressure level, for this condition is approximately 99 db.

Fig. 4-4 and 4-5 show similar spectra for operating conditions where entrained air exists at the pump inlet due to low inlet pressure and injected air. When aeration is caused by low inlet pressure, the "all-pass" level is 100 db. When aeration is induced by injecting air, the "all-pass" level was 99 db. Both of these spectra were obtained after the air-oil mixture at the inlet had reached a "steady-state" or homogeneous condition. These data imply that entrained air which is thoroughly mixed with the oil has little effect on the overall sound level of a hydraulic pump.

some combination of K_1 , α , and β can be used to establish acceptable sound power performance limits for fluid power pumps.

PUMP SOUND VS. % AIR

Fluid power systems frequently operate with entrained air in the fluid. This undesirable condition may exist because of air ingestion due to wear or loose fittings. In some cases, aeration exists due to faulty design. The question arises as to what the entrained air does to the pump sound level. Fig. 4-3 shows the frequency spectrum for a pump operating under "normal" conditions or a condition where

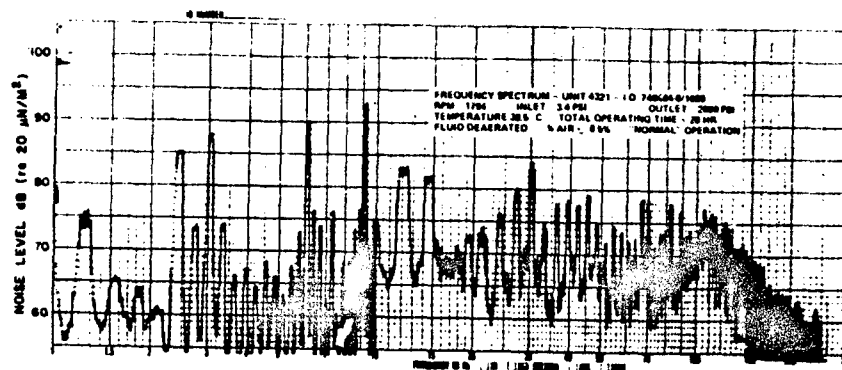


Fig. 4-3. Frequency Spectrum for "Normal" Operating Conditions.

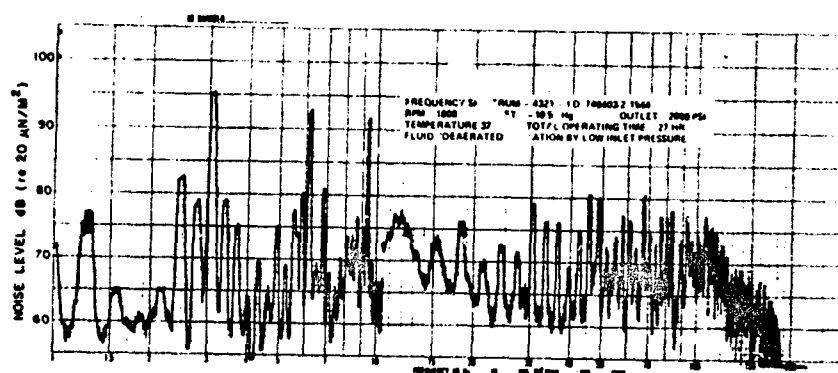


Fig. 4-4. Frequency Spectrum for Pump Operating with Inlet Aeration Due to Low Inlet Pressure.

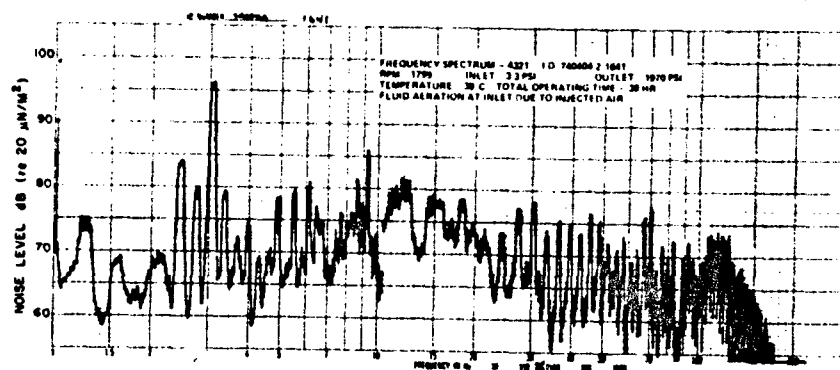


Fig. 4-5. Frequency Spectrum for Pump Operating with Inlet Aeration Due to Injected Air.

The condition of aeration which does cause noticeable changes in a pump's sound power output occurs when small volumes, or pockets, of air that are not homogeneously mixed with the oil pass through the pump. Since this is a random process, it is difficult to control for test purposes, but very noticeable sound changes were noted while the air-oil mixtures were stabilizing for the tests of Figs. 4-4 and 4-5. The point is that homogeneous mixtures of air-oil may change the character of a pump's sound but do not appear to significantly affect the overall sound level. If "pockets" of air are introduced into the oil and allowed to reach the pump inlet, the resultant sound fluctuations will be greater than a db or two and will occur frequently enough to be annoying.

If a hydraulic system is operated independently on a machine and the engine is surging while the pump frequently sounds like "*it is grinding marbles*," it is highly probable that the system is allowing an air-oil two-phase flow to reach the pump inlet.

PUMP ABN ISOLATION

Two techniques for isolating directly radiated pump abn were studied for this report. The first technique was simply building an acoustical isolator. The second technique was putting damping compound on the pump in an attempt to dissipate the vibrational energy as heat in the damping compound.

Several abn noise isolation techniques were used. The most successful reduced the sound level 12 dbA. However, none of the treatments used in the laboratory were considered practical for machine applications. The combinations of materials were similar to those described in the chapter on conduit noise. Basically, they consisted of de-coupling the pump from an acoustical barrier material with "*foam*" materials and increasing the barrier material and associated absorption material to achieve the sound level reduction.

Table 4-3 summarizes the results of tests using damping materials to modify the sound power emitted by pumps numbers 36 and 37. Although the results at some

**TABLE 4-3. TEST RESULTS USING DAMPING MATERIALS ON PUMPS TO REDUCE ABN.
SOUND POWER dbA.**

Pump	Test Condition	Plain	1/8" Sound Off	1/8" Sound Off + 1/8" Ceramic	1/8" IN170306 Ceramic	3/4" Ceramic
36	2000, 2000	96.7	95.5	96.5	----	----
	2000, 200	81.6	84.4	88.4	----	----
	600, 1800	79.6	76.8	82.0	----	----
	600, 200	63.0	65.5	63.3	----	----
37	2000, 2000	94.1	---	---	97.4	97.3
	2000, 200	78.8	---	---	-----	89.7
	600, 2000	75.7	---	---	77.6	75.4
	600, 200	66.0	---	---	---	61.9

Test Data I.D. 740619

speeds and pressures appear encouraging. the overall results did not indicate that direct application to the pump's surface was a very practical approach to reducing pump abn. . It should be emphasized that the treatment thickness was probably less than the recommended 1.75 times the thickness of the surface being treated. However, after putting as much as 3/4 of an inch of damping compound on pump 37 and finding the sound level to be over 10 dbA higher, it was decided to terminate the testing. The continued addition of damping compound apparently increased the surface area enough to make the combination of pump and compound a more efficient radiator of certain pumping frequencies, in spite of the additional damping.

PUMP SBN ISOLATION

Test results using pump sbn isolation were very encouraging. Fig. 4-6 shows the basic vibration isolation technique that was used to decouple the pump from the pump mount. The pump pilot diameter, concentric with the shaft, was not allowed to touch

the mount when isolators were used at the mounting bolts.

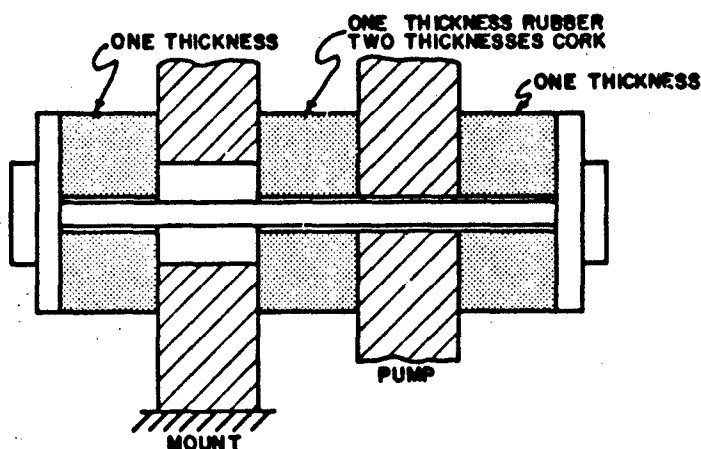


Fig. 4-3. Cross-Section of Pump Isolation Technique.

The test results are summarized in Table 4-4. As shown by the data, the overall vibrational level of the pump surface was not significantly altered by either isolation technique. However, both the cork and rubber isolators reduced the vibrational level of the mount by 25 db. The rubber mounts appeared to isolate the vibration in the "audible" frequency range better than the cork mounts,

as evidenced by the levels at the 1/3 octave frequency of 1250 Hz.

The isolators used for these tests were fabricated at the FPRC using available materials. It is reasonable to believe that commercially available bolt isolators could be used effectively if the pump pilot diameter was isolated from the mount.

TABLE 4-4. SUMMARY OF SBN ISOLATION TESTS.

COUPLING	PUMP (ALL-PASS)	MOUNT (ALL-PASS)	MOUNT (1250 Hz.)
Direct	89.0	114.0	88.0
Cork	88.5	91.0	79.0
Rubber	90.0	91.0	84.5

Velocity, db, 741122-1, 2, & 3. (100db, 1g, 100 cps).

CHAPTER V

CONDUIT NOISE

Conduit noise in fluid power systems can be traced to hydraulic system pressure ripple and structureborne vibrations. This interaction is depicted in Fig. 5-1. When it is not possible to reduce the fluidborne noise or the structural vibrations, conduit-radiated noise is frequently reduced with acoustical isolators. This chapter presents the results of a study of two acoustical isolation materials for hydraulic system conduits.

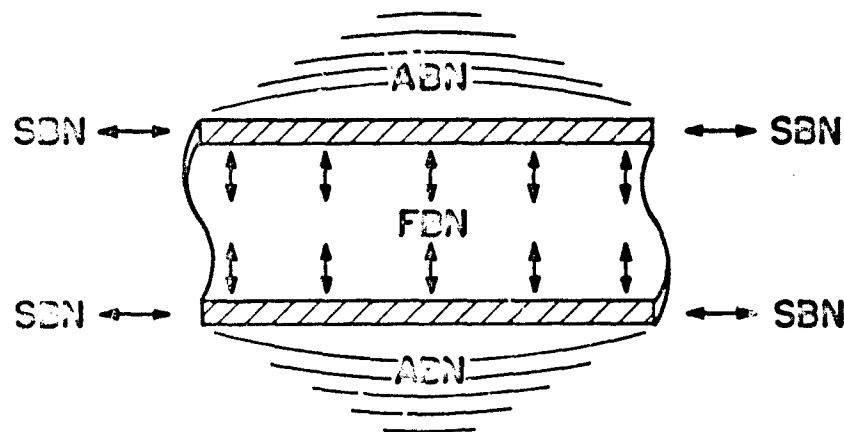


Fig. 5-1. Illustration of Two Sources of Conduit Noise — Vibration and Pressure Ripple.

The tests were conducted in the FPRC Reverberant Facility. A ten-foot section of one-inch steel conduit was isolated in the reverberant chamber with flow being introduced to the conduit through piping embedded in a large concrete mass. The mass, which is isolated from the embedded pipes with foam, minimizes structurally induced conduit vibration. Any resultant vibration of the test specimen is then due to flow-induced vibration and fluidborne noise.

The two treatments used for the tests were: (1) a commercially available material called "Pipe Wrap" and (2) an experimentally fabricated combination of two-inch foam and commercially available leaded vinyl. The "Pipe Wrap" consists of a one-inch layer of foam rubber that is covered by sheets of lead and aluminum that are elastically bonded together to provide constrained layer damping. The second isolation material is the wrapping technique used by the FPRC during acoustical airborne acoustical evaluation of fluid power components to reduce the airborne noise radiated from conduits connected to test specimens in the reverberant facility.

The variables during testing were the pump speed and pressure, the isolation materials, the manner in which the materials were installed on the conduit, and the installation of a fluidborne noise attenuator in the fluid system (outside of the test facility).

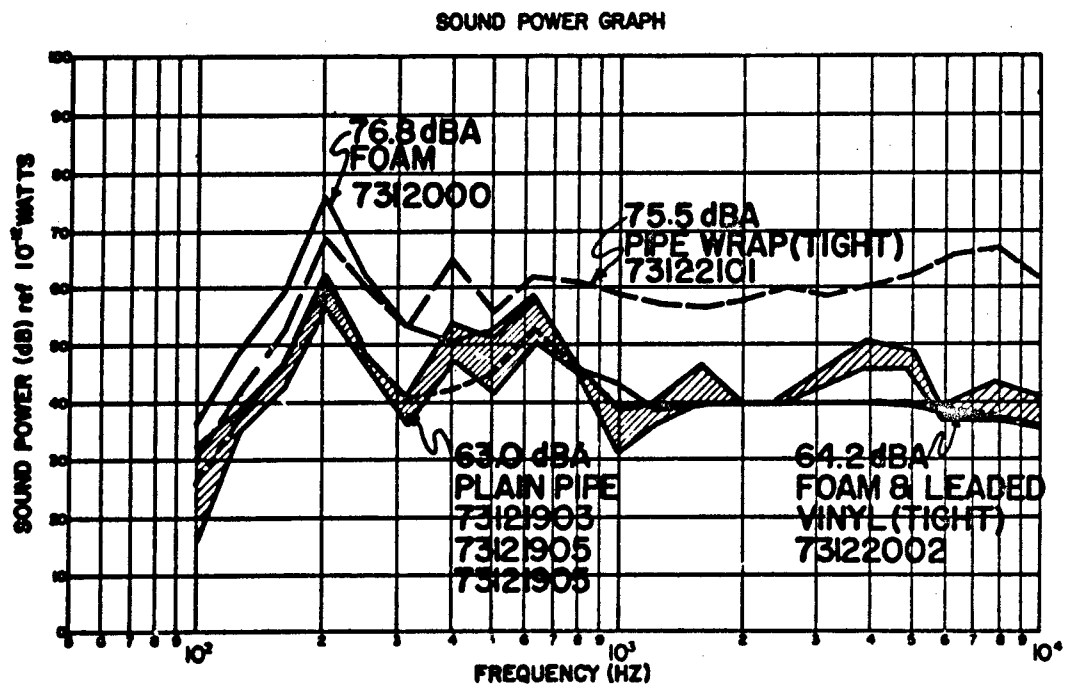


Fig. 8-2. Sound Power Vs. Frequency for Conduit with Pulseco Installed Downstream of Conduit. Pump Speed 1500 rpm, Outlet Pressure 2000 psi.

Fig. 5-2 compares the overall dbA sound power levels for various conduit treatments when the pump speed was 1500 rpm, the pressure 2000 psi, and the isolation materials "tightly" wrapped on the conduit. For these tests, a "Pulsco," fbn attenuator, was installed downstream of the conduit. These tests implied that the isolation materials would not reduce the dbA sound power of the conduit. A similar comparison – pump speed 2000 rpm, pressure 2000 psi – also indicated that the isolation wrappings only tended to act as more efficient radiators of the structural vibration in the conduit (Fig. 5-3).

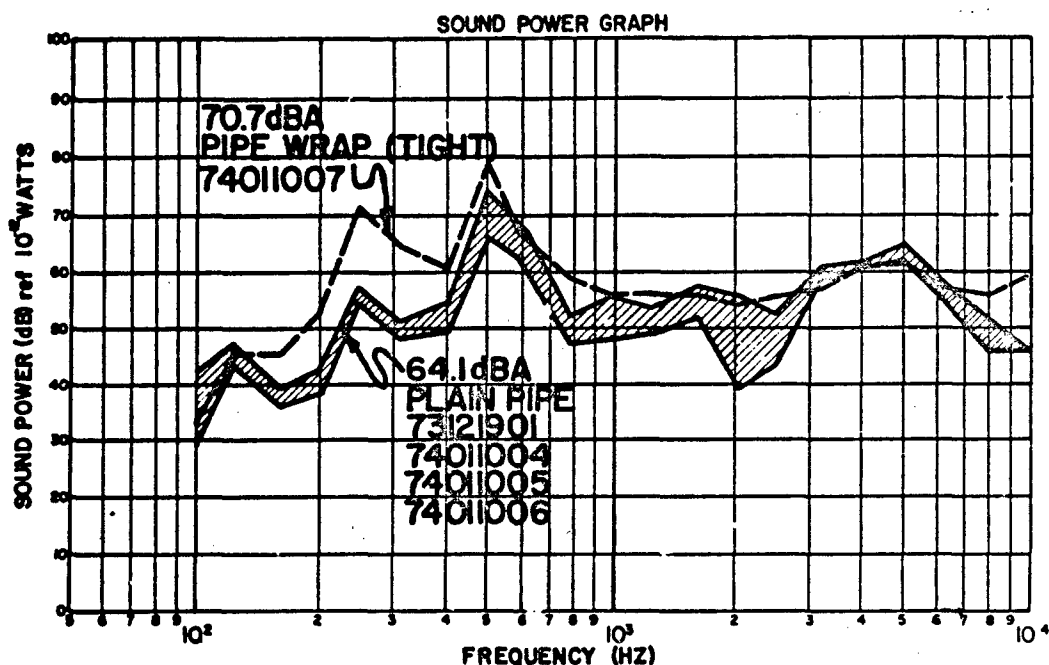


Fig. 5-3. Sound Power Vs. Frequency for Conduit with Pulsco Installed Downstream of Conduit. Pump Speed 2000 rpm, Pressure 2000 psi.

In order to determine if the *tight* wrappings had indeed cancelled the effectiveness of the isolation materials, further tests were conducted with the isolation materials "loosely" installed on the conduit. The "Pulsco" was also removed from the circuit because the flow rate for most of the tests was above the rated flow for the attenuator. The test results shown in Fig. 5-4 indicated that it is important to install the isolation

materials so that they are properly "de-coupled" from the conduit. Figs. 5-5 and 5-6 also indicate that, if the isolation materials are installed to take advantage of their sound transmission loss, then the noise from the conduit will be significantly reduced.

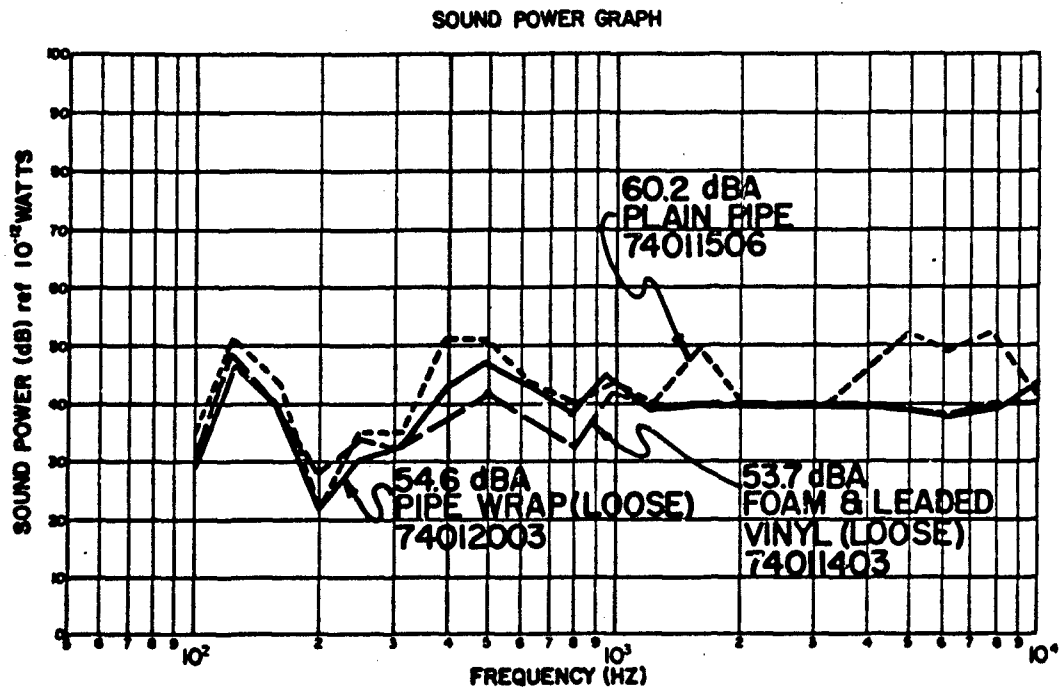


Fig. 5-4. Sound Power Vs. Frequency for Conduit without Pulsco Installed. Pump Speed 1000 rpm, Pressure 2000 psi.

It should be pointed out that the hydraulic power supply for these tests was located about 40 feet from the test conduit. This distance between the supply pump coupled with several pipe diameter changes between the pump and the test conduit reduced the pump pressure ripple appreciably before the hydraulic fluid entered the test specimen. Therefore, the sound levels shown in the test results are lower than those normally encountered for hydraulic conduits. The sound levels are still adequate for the evaluation of the isolation materials.

More than 30 different tests were conducted on the conduit to evaluate the effects of the "Pulsco" and the two different isolation materials. The more significant results

of those tests are summarized in Table 5-1. A review of this table shows that the two-inch foam and leaded vinyl combination is the most effective acoustical isolator. Both the "Pipe Wrap" and the "Pulsco" reduce the conduit's apparent sound power, but neither is as effective as the foam leaded vinyl combination. It is certainly apparent that conduit acoustical isolation can be installed so tightly that the material becomes an amplifier rather than an attenuator.

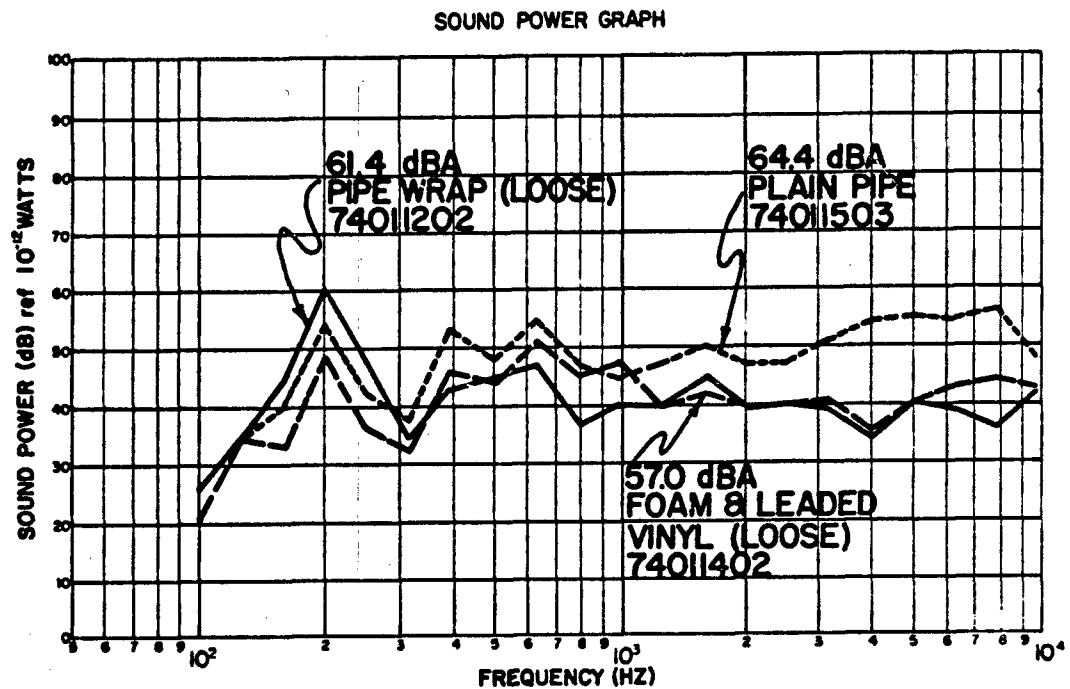


Fig. 5-5. Sound Power Vs. Frequency for Conduit without Pulsco Installed. Pump Speed 1500 rpm, Pressure 2000 psi.

TABLE 5-1. AIRBORNE CONDUIT SOUND POWER db/dbA AS A FUNCTION OF ABN ISOLATOR TREATMENT.

TEST CONDITIONS	TREATMENT							
	Foam Fulseco Down- stream	Foam + Leaded Vinyl (Tight) Fulseco Down- stream	Foam + Leaded Vinyl (Loose) No Fulseco	Plain Conduit Fulseco Down- stream	Plain Conduit No Fulseco	Plain Conduit Fulseco Upstream	Pipe Wrap (Loose) No Fulseco	Pipe Wrap (Tight) Fulseco Down- stream
1000 rpm * ~ 12.5 gpm 2000 psi			65.5/53.7 ==	62.1/58.0	68.2/60.2	63.1/57.3	63.2/54.6 ==	
1500 rpm ~ 25.2 gpm 2000 psi	** 87.5/76.5 87.5/76.5	76.0/64.2	62.4/57.0 ==	70.6/61.6 73.2/64.0 71.5/63.4	68.1/64.4	63.0/61.9	71.6/61.4	81.7/75.5
2000 rpm ~ 32 gpm 2000 psi			63.4/60.6 ==	68.9/66.6 66.8/64.1 66.8/64.0 64.4/61.8	69.0/68.0	67.0/66.6	67.4/64.2	75.0/70.7
2000 rpm 1000 psi				59.6/56.0 63.8/58.3 64.7/57.3				75.8/75.5
2000 rpm 750 psi								76.4/72.9

*Recommended flow for Fulseco-Apt-4 is 0-14 gpm.

**The numbers shown in the chart, such as 87.5/76.5 mean 87.5 db/76.5 dbA.

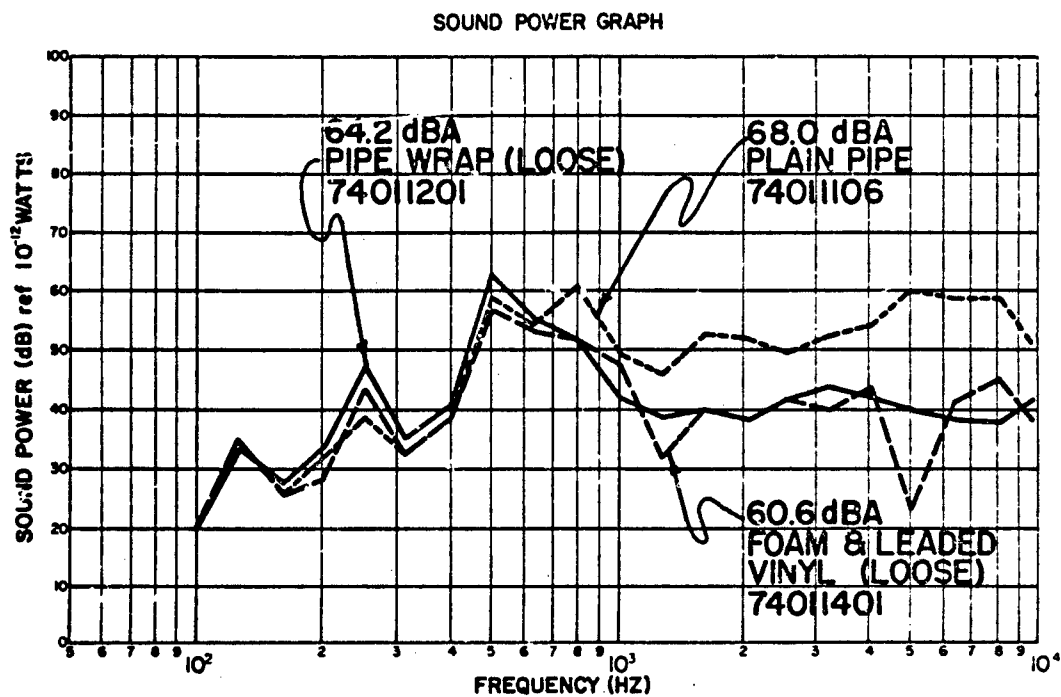


Fig. 5-6. Sound Power Vs. Frequency for Conduit without Pulsco Installed. Pump Speed 2000 rpm, Pressure 2000 psi.

CHAPTER VI

RESERVOIR NOISE

Hydraulic reservoir noise is caused by sbn and fbn introduced to the reservoir by the hydraulic system and the machine frame. Two readily available noise controllers for reservoirs are vibration isolators and damping compound. This chapter discusses the results of reservoir noise studies using both techniques to reduce airborne noise emitted by hydraulic reservoirs.

STRUCTUREBORNE NOISE

In order to study the effects of structureborne noise attenuators and isolators, a reservoir from the 6000 lb. rough terrain fork lift was mounted on a vibration test fixture in the FPRC Reverberant Facility. The vibration test fixture (see Fig. 6-1) was constructed with an input arm which was connected to a shaker. The shaker transmitted energy to the test fixture, which transmitted the energy to the reservoir. The test fixture was structurally isolated from the floor of the reverberant facility. The test procedure was to measure airborne noise in the facility while exciting the test fixture with a known amplitude from the shaker. It was established that the background noise level was significantly below the noise level of the untreated reservoir. The background level was due to the shaker and its power supply, which were installed in the test facility.

The test sequence was as follows:

1. Establish the mount resonant frequency for the untreated reservoir. Measure the sound power level in the room and record the input shaker displacement.
2. Isolate the reservoir from the test fixture with vibration isolators and record the sound power level for the established test condition.

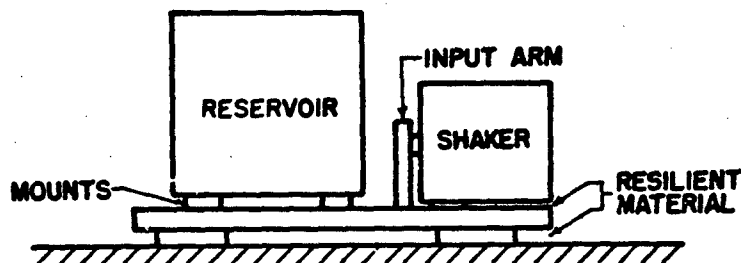


Fig. 6-1. Vibration Test Fixture.

For all of these tests, the reservoir was filled with hydraulic oil. The damping compound was IN170306. The vibration isolators were Barry Number C-2150.

The results of the tests on the fork lift reservoir are summarized in Table 6-1. It is interesting to note that the combination of both the damping compound and the isolators was not as effective as the isolators alone. Since each of the reductions using the damping compound and isolators was very effective (7 to 19 db drop), it seems apparent that these techniques offer a workable solution to the problem of reservoir noise.

TABLE 6-1. SUMMARY OF STRUCTUREBORNE NOISE CONTROL STUDY ON 6000 LB RTFL RESERVOIR, SOUND POWER LEVEL (db).

STRUCTURAL ISOLATION	DAMPING COMPOUND	250 HZ * (1/3 OCTAVE)	db CHANGE
NONE	NONE	89	-0
NONE	YES	82	-7
YES	YES	77	-12
YES	NONE	70	-19

*250 Hz. was determined to be the natural frequency of the reservoir.

3. Cover the reservoir with damping compound and with the vibration isolators still installed record the sound power at the test condition.
4. Remove the isolators and measure the sound power at the test condition while the reservoir was still covered with damping compound.

The fact that the combination of isolation mounts and damping compound did not give the greatest noise reduction compared to the untreated reservoir prompted another set of tests on a smaller reservoir using a different damping compound, Kinetics KDC-E-162.

The reservoir for the second set of tests was about one-half the volume of the 6000 lb RTFL reservoir. Another test fixture was constructed. The second test fixture was driven in the same manner as the first. The test procedure for the second set of reservoir tests was similar to that used for the first reservoir, except that the number of data points was increased. For the smaller reservoir, data were taken at each of the 1/3 octave center frequencies from 100 Hz. to 1000 Hz. Although this increased the data acquisition and reduction effort, the results offered a broader view of the performance of the damping compound and the vibration isolators.

The results of the second set of tests are summarized in Fig. 6-2 and Table 6-2. These results are consistent with the first tests in that the isolation mounts are shown to be the best way to reduce reservoir noise due to frame-induced vibration. The vibration isolators do not allow the cause of the noise to reach the reservoir. Once again, the combination of both isolators and damping compound is not the most effective for reducing noise. These results may lead to the erroneous conclusion that damping compound should not be used for reservoir noise control.

TABLE 6-2. SUMMARY OF SOUND POWER DATA FOR SMALL RESERVOIR. INPUT DISPLACEMENT CONSTANT.

Structural Isolation	Damping Compound	SOUND POWER			dba CHANGE
		500 Hz (db)	100 Hz — 1000 Hz (db)	100 Hz 1000 Hz (dBA)	
NONE	NONE	86	90	87	0
YES	YES	67	74	70	-17
NONE	YES	63	71	69	-18
YES	NONE	63	71	67	-20

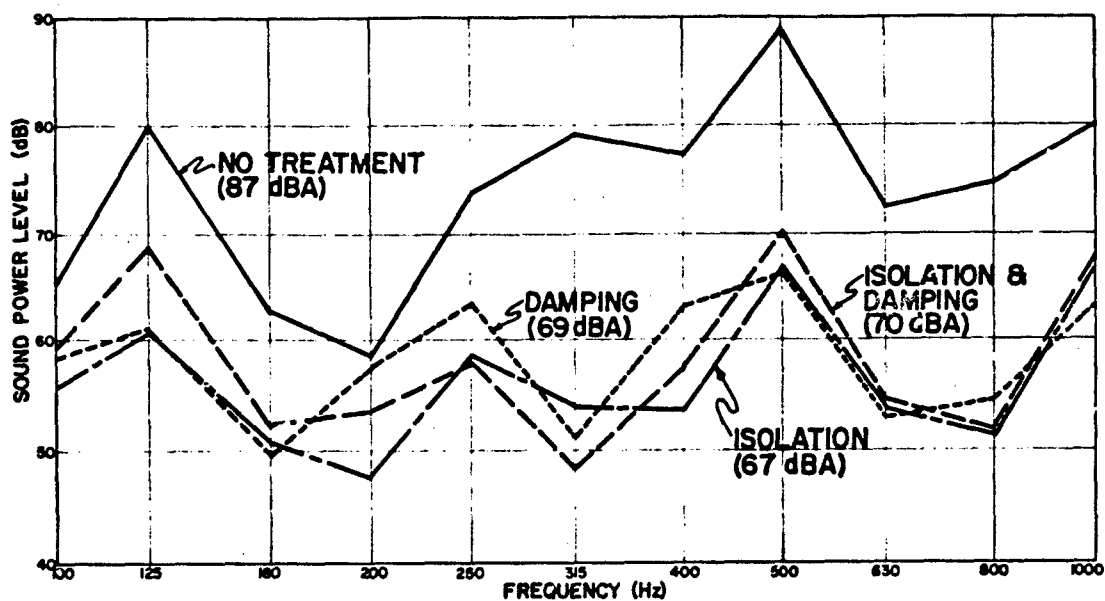


Fig. 6-2. Effects of Reservoir Noise Reduction Techniques Applied to Small Reservoir. Input Displacement Constant dbA Levels Are Summed Between 100 Hz and 1000 Hz.

One source of reservoir vibration is the array of hydraulic conduits conveying fluid to and from the reservoir. Since it is not feasible at this time to structurally isolate the conduits at the reservoir surface (no components are readily available for doing this), damping compound offers a practical approach to minimizing the effects of conduit vibration. Considering yet another source of reservoir vibration, the damping compound offers a technique for minimizing any adverse effects due to pressure ripple.

FLUIDBORNE NOISE

A reservoir from a 6000 lb. rough terrain fork lift was isolated in the FPRC Reverberant Facility and connected to a hydraulic system which was to act as a forcing function. The conduits connected to the reservoir were wrapped to isolate any airborne noise that they might generate. The conduits entered the test environment in such a way that conduit structureborne noise transmission from outside of the test facility was minimized. For the test, pressure ripple in the hydraulic system was the primary forcing function. Testing

revealed that there was insufficient pressure ripple reaching the reservoir. The sound level with the system operating was so close to the background level of the facility (less than 55 dbA sound power) that it would have been impossible to accurately assess the noise reduction capability of any damping compound.

Some consideration was given to introducing a pump into the test environment. This would have added another major noise source to the environment. Previous work, measuring "hydraulic speaker" noise with a pump in the test environment, had emphasized the difficulty of isolating pump noise in order to measure another source in the environment.

In order to obtain some indication of how effective damping compound would be on a reservoir being forced by structural vibration and fluid pulsations, a noise reduction study was conducted on a small hydraulic test stand. The results of the study are shown in Fig. 6-3. At the maximum speed tested, there was about 10^{-3} watts at the test position for the untreated stand. After adding damping compound to the reservoir, the sound power at the test position dropped to $0.5 \cdot 10^{-3}$ watts. This means that the reservoir produced between 87 and 90 dbA sound pressure at the test location. Treating the pump reduced the power at the test location to $0.33 \cdot 10^{-3}$ watts, which means that the pump was producing between 82 and 87 dbA sound pressure at the test location.

In other words, treating the reservoir was like removing a source equivalent to 87 dbA at the test position, while treating the pump was only comparable to removing a source of 82 dbA. It would have been extremely difficult to quiet the test stand without treating the reservoir with damping compound or isolating the reservoir with acoustical isolation materials. Compared to "lagging" materials, damping compound is easily applied to irregular surfaces. Damping compound has proven to be durable in a hydraulic environment.

The 6000 lb. rough terrain fork lift reservoir that was initially treated with a ceramic like damping compound (IMH No. 70306) was placed outside to "weather." After several months of exposure to sun, rain, and snow, the treatment is still adhering to the reservoir surface. This indicates that damping compound can be reasonably durable. More conclusive durability tests should be conducted before the materials are used extensively in the field.

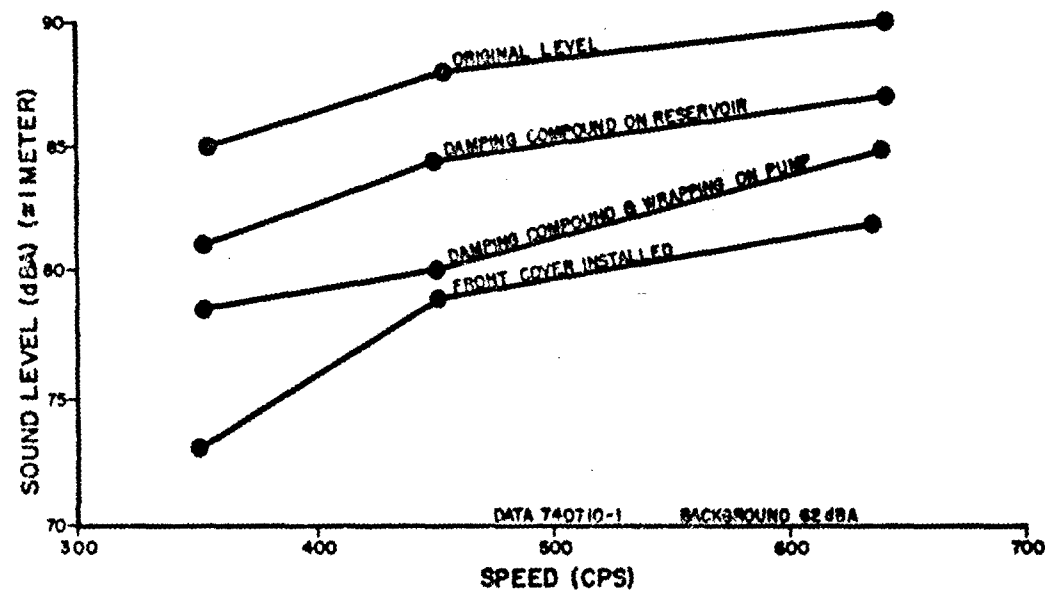


Fig. 6-3. Sound Level Changes of Filter Test Stand Due to Modifications.

CHAPTER VII

FLUIDBORNE NOISE CONTROL

It has been emphasized that fluidborne noise, pressure ripple, is the one characteristic of hydraulic systems that — acoustically speaking — makes them unique from other systems on modern machines. On many hydraulically operated machines, one of the primary sources of vibration is fluid pressure ripple. If it is not possible to obtain pumps that have low pressure ripple, which would be selecting a "quiet" source, then the designer may be forced to consider attenuating the noise in the transmission path. The latter course of action means using some sort of fluidborne noise attenuator. There are two types of fluidborne noise attenuators commercially available. The first type is basically an accumulator; the second type is an acoustic filter (similar to a muffler). Both of these types of pressure ripple attenuators were examined for this study.

PULSCO

The PULSCO is a reactive type pulsation attenuator which performs much like an engine exhaust muffler. The PULSCO contains no moving parts. The unit tested for this study was an APT-4, which has 3/4 inch pipe ports. The APT-4 is rated for a maximum flow rate of 14 gallons per minute. For some of the tests reported in this chapter, the flow through the unit exceeded the rated flow. The unit has approximately 100 psi pressure drop at 14 gallons per minute. The PULSCO weighs about 41 pounds and is rated for 3000 psi operating pressure.

The data in Table 5-1 show how the PULSCO affected the noise emitted by a piece of conduit in the OSU-FPRC Reverberant Facility. In reviewing the data in Table 5-1, it can be seen that, whether the PULSCO was upstream or downstream of the conduit, it acted to reduce the radiated airborne noise. Table 7-1 summarizes several sets of test data

TABLE 7-1. OVERALL CHANGES IN dbA LEVELS WITH PULSCO INSTALLED.

		PULSCO Upstream	No PULSCO	PULSCO Downstream
1000 rpm 2000 psi	ADN	57.3 dbA	60.2 dbA	58.0 dbA
	FBN	186.0 dbA	194.5 dbA	*
1500 rpm 2000 psi	ADN	61.9 dbA	64.4 dbA	63.0 dbA
	FBN	*	204.2 dbA	210.5 dbA

*Data not reduced. Data sets 731219-2, 4, 6, 740111-1, 47, 740115-1, 4, 8, 12, 740116-2.

in terms of dbA sound power from the conduit and dbA pressure ripple in the conduit. It is interesting to note that, in all tests with the PULSCO installed, the sound from the conduit is reduced, even though in one case the pressure ripple is greater. These results are reflected in Figs. 7-1 and 7-2.

Fig. 7-1 shows the fluidborne noise levels in the test conduit with no PULSCO, the PULSCO upstream, and the PULSCO installed downstream of the conduit. It is reasonable to believe that the upstream noise source is the pump and that the downstream source is the load valve. Since the pump produces noise at the fundamental pumping frequency and its harmonics, while the valve tends to be a higher frequency broad-band source, one might expect the PULSCO, when installed upstream, to reduce the low frequency noise and, when installed downstream, to reduce the higher frequency noise. Indeed, this hypothesis seems to explain the behavior demonstrated by the curves of Fig. 7-1, where the PULSCO appears to amplify the low frequency noise and attenuate the high frequency noise when installed downstream. When the PULSCO is installed upstream, it appears to reduce the primary pumping frequencies and may even slightly amplify the higher frequency ripple. These same trends occur in the sound power radiated from the conduit, as shown in Fig. 7-2.

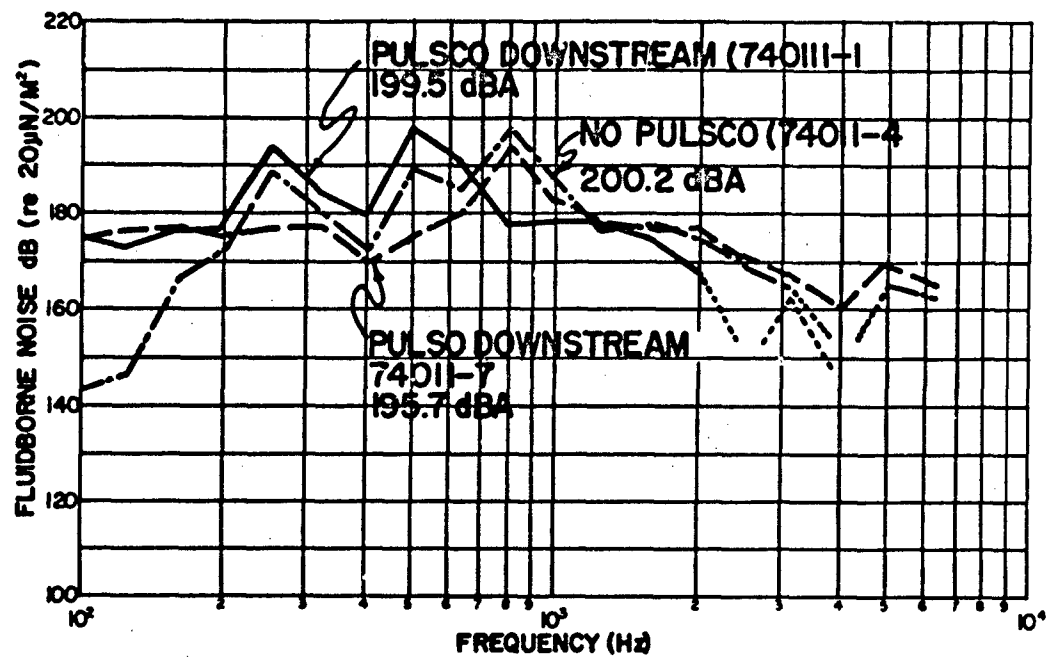


Fig. 7-1. Fluidborne Noise Levels in Conduit for Various Test Conditions with PULSCO. Pump NP-1, 2000 rpm, 2000 psi.

The data shown in Figs. 7-1 and 7-2 can be used to discuss an insertion loss for the PULSCO. Insertion loss is the difference between the level without the attenuator and the level with the attenuator. In terms of fluidborne noise, if measurements are made upstream of the attenuator and downstream of the attenuator simultaneously, then the difference between the upstream level and the downstream level is referred to as a transmission loss.

In order to obtain comparative results for two types of FBN attenuators, a PULSCO and a Pulse-Tone were tested for transmission loss under the same conditions, in the same system, with the same pump, using equal line lengths. The only difference between the tests was the substitution of one unit for the other. The results of the experiments are shown in Fig. 7-3.

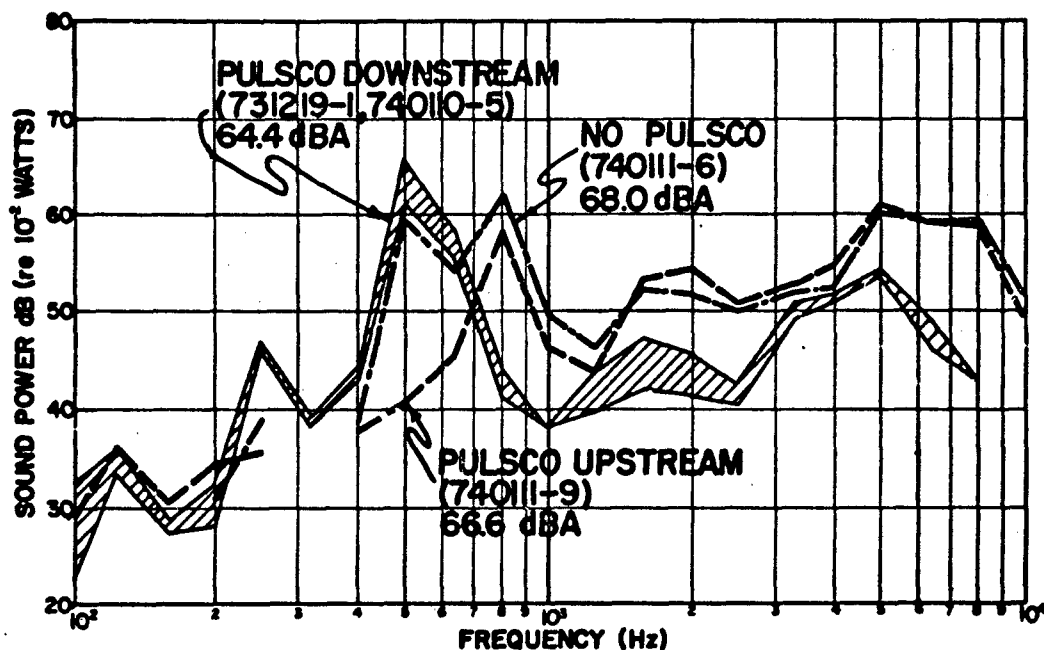
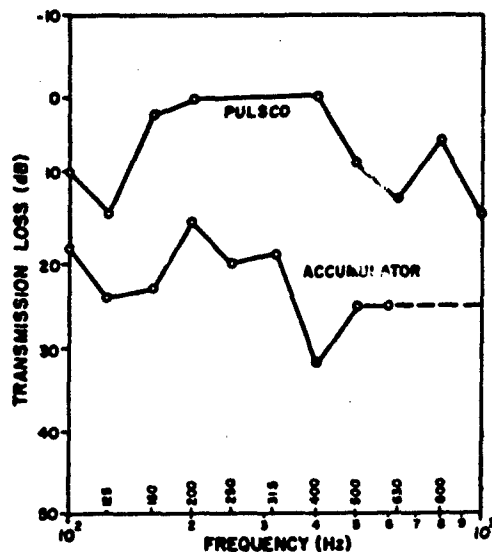


Fig. 7-2. Conduit Sound Power for Various Test Conditions with PULSCO. Pump NP-1, 2000 rpm, 2000 psi.

PULSE-TONE

The accumulator test results shown in Fig. 7-3 were obtained with a Greer Pulse-Tone. The Pulse-Tone is a 15 in.³ accumulator which weighs about seven pounds. The unit is rated for 2000 psi operating pressure and has about 15 psi pressure drop at 14 gallons per minute flow rate with Mil-2104 at 120°F. Fig. 7-4 shows the results of another typical test with the Pulse-Tone. The performance of the Pulse-Tone is well documented in Refs. [7] and [8]. Fig. 7-5 shows an insertion loss prediction curve for the Pulse-Tone.

Fig. 7-6 shows a published [9] insertion loss prediction curve for the PULSCO. The data points on the curve are maximum attenuation points from test data for this study. The results of the tests for this study and the results of other studies of FBN attenuators



System Pump = OSU NP-1, Type = External Gear
 Flow Rate = 15 GPM Speed = 1000 RPM
 System Pressure = 2000 psi
 Accumulator Volume = 15 in.³
 Accumulator Precharge = 1000 psi
 Pulsco Type = Apt-4

Fig. 7-2. Comparison of the FBN Transmission Loss Obtained with Two Types of FBN Attenuators.

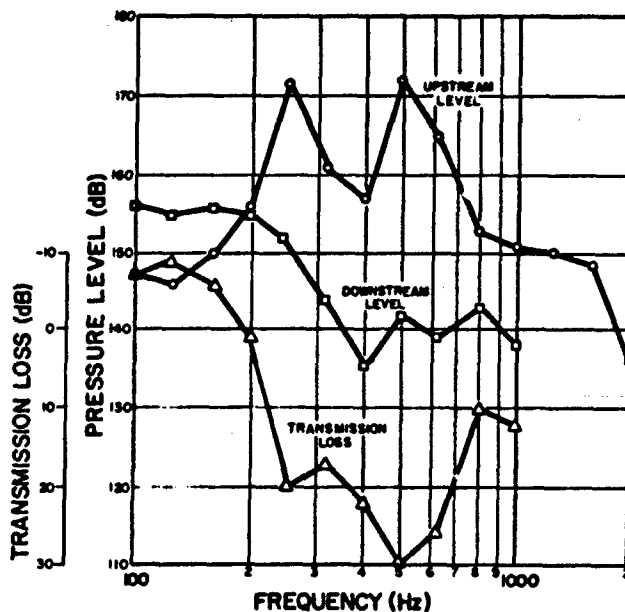


Fig. 7-4. Data for 2000 rpm Pump Speed, 15 in.³ Pulse-Tone, 2000 psi System Pressure, 1000 psi Precharge.

indicate that they do not simultaneously attenuate the FBN at all frequencies as suggested in Figs. 7-5 and 7-6. Ref. [7] contains a complete discussion of this nonlinear characteristic of FBN attenuators. The designer wishing to use either Fig. 7-5 or 7-6 for predicting purposes is encouraged to study Ref. [7].

Pumping Element Orientation

The use of fluidborne noise attenuators, such as the PULSCO or the Pulse-Tone, can be an effective means of controlling hydraulic pressure ripple. When possible, it is better to modify the source to reduce FBN. Inlet pressure ripple can be a significant problem, since inlet line vibrations also contribute to the overall system noise level. On dual outlet pumps with a common inlet, one possible approach to minimizing the inlet pressure ripple would be to orient the pumping elements so that they are out of phase. Thus, instead

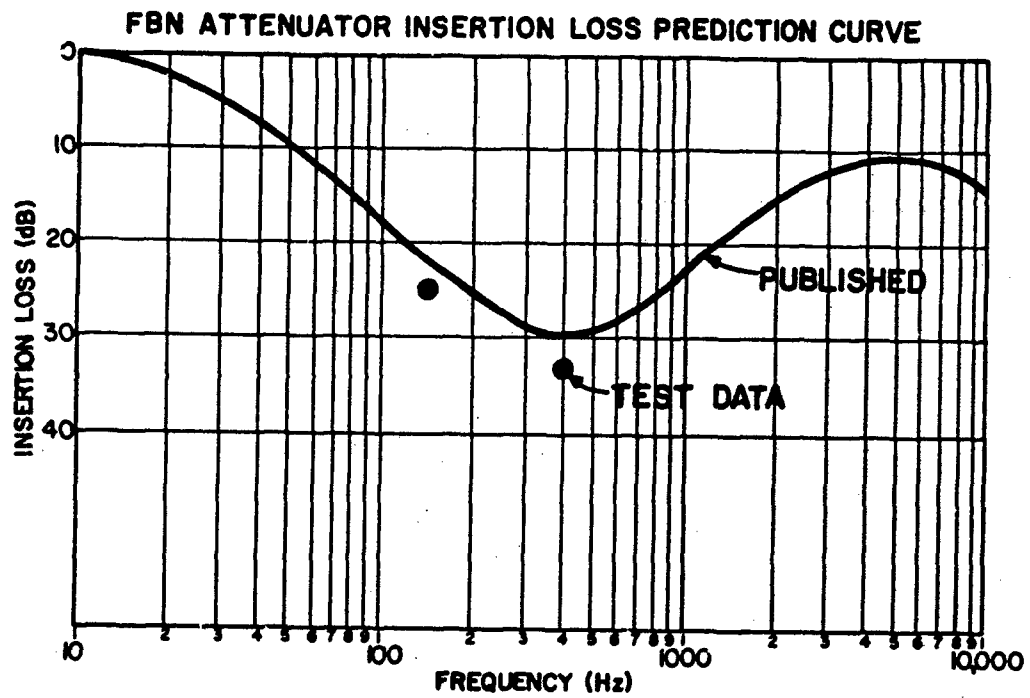


Fig. 7-5. Pulse-Tone Fluidborne Noise Controller Evaluation Results [7].

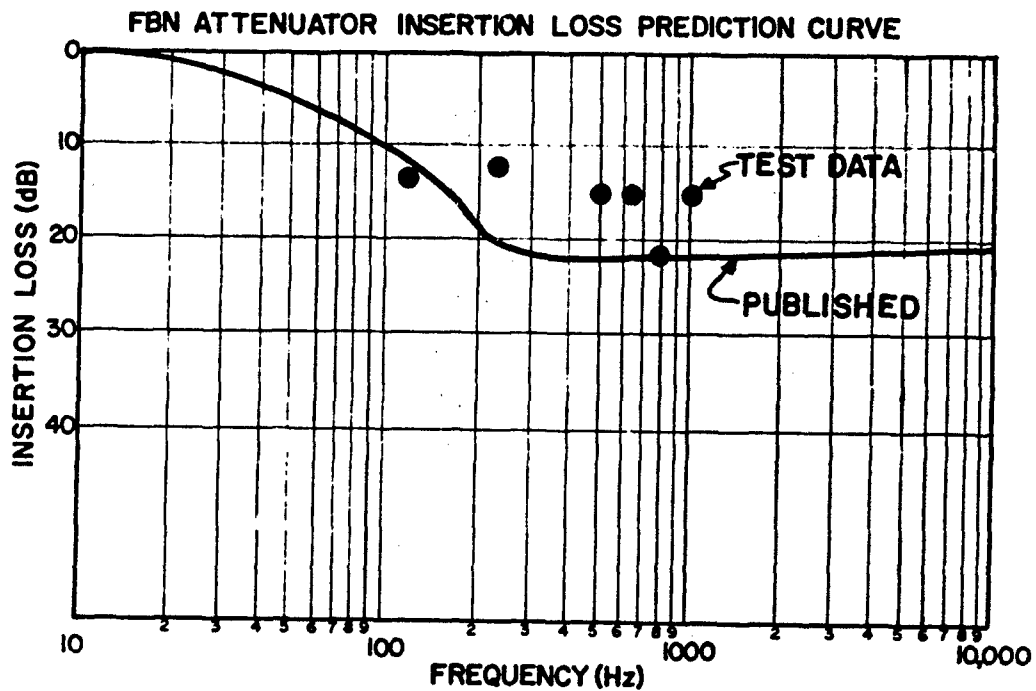


Fig. 7-6. PULSCO FBN Controller Published Characteristics and Test Results [9].

of the inlet flows reinforcing one another to produce large inlet flow ripples, the gears or vanes could be adjusted to minimize the inlet flow ripples.

CHAPTER VIII

SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS

This report presents a technique for assessing the noise associated with fluid power systems, shows the sound power characteristics of several components from a 6000 lb. rough terrain fork lift, and summarizes the results of tests that show the feasibility of several noise reduction techniques for fluid power systems. The volumes of test results were summarized in order to succinctly present the major effects of the various noise control methods. Although it is beyond the scope of this report, it is possible to more fully explain the behavior of most of the noise control techniques using linear theory.

A review of linear theory can be found in many vibration control and acoustics books. Although linear theory is a powerful tool for estimating the response of simple systems and a very useful tool for communicating concepts, the reader should not be shocked if the results of these studies do not always follow basic linear theory. The test results in this report represent the response of real components, operating as they would in real systems. This type of testing brings out many of the nonlinearities in the "real" world, which are sometimes difficult to explain. But, the design engineer can be reasonably confident that, if it worked during the tests for this study, it can be adapted to control noise in the field.

The system noise model presented in Chapter II provides an accounting approach to estimating the noise associated with a given fluid power system. The tabular model outlines the "*pieces of information*" that are needed to fully assess fluid power system noise. Careful evaluation of the tabular model will yield an insight for the engineer who wishes to quiet a fluid power system. The model can only become more powerful if new information is made available. More complete information will become available when the industry develops the necessary test procedures and subsequently makes the data available.

It is emphasized that there are only three ways to quiet a system — change the operating conditions, change components, or add noise controllers.

A pump noise model is introduced in Chapter IV. The model is a good estimator of the modeled pump's sound power over a portion of its operating region. Reviewing sound power data on the pumps from the 6000 rough terrain fork lift shows that, for pumps 30, 31, and 32, the sample standard deviation is approximately 1.0 dbA. For pumps 36 and 37, an estimate of the sample standard deviation is 2.3 dbA. Pumps 36 and 37 contain a valve which may have contributed significantly to the overall unit sound power. The use of damping compound on pumps as a noise control technique shows little promise for small pumps. However, it definitely appears that some type of structural isolation between hydraulic pumps and the frame (or mount) offers an excellent means of reducing the energy transferred between the machine and the pump.

Although the results of Chapter V show that conduit noise can be isolated with covering materials, it would appear initially that attacking the source — FBN — is a more practical approach. Pipe coverings which are very practical in permanent stationary installations may encounter durability problems on mobile equipment and create maintenance difficulties.

Both vibration isolators and damping compound contribute significantly to the reduction of reservoir noise. Since both products are durable and both help contain or reduce the total system energy, they should be seriously considered for field use.

The two important parameters for evaluating fluidborne noise controllers are pressure drop and insertion loss (transmission loss). Ideally, the input and output impedance of the pressure ripple attenuators should be defined in order to provide the proper information for total system analysis. But, for the present, the proper test procedure, based on the insertion loss of an attenuator, could give adequate information for designers to discriminate between components.

The results of this study show that careful component selection, fluidborne noise attenuators, damping compound, and vibration isolators can play an effective role in reducing fluid power system noise.

It is noted in Appendix A that, as a part of this study, project members participated in the development of an ISO test procedure for measuring sound power emitted by fluid power pumps. This procedure will give fluid power system designers a means of obtaining data which can be used to acoustically discriminate between components being considered for the same application.

The military can obtain "quiet" machines by specifying "quiet" vehicles, which implies specifying "quiet" systems and "quiet" components, and by employing noise reduction techniques to retrofit existing machines. The following recommendations are oriented toward implementing those options:

1. Manufacturers should be encouraged to produce practical vibration isolators for hydraulic pumps and motors.
2. Test procedures should be developed and utilized to specify components on an acoustical basis:
 - A. FBN attenuatorsInsertion Loss
 - B. FBN pumps.....Measure a pump's potential to produce system noise.
 - C. ABN valves.....Measure the sound power of valves.
3. Using available test procedures for hydraulic pump and motor sound measurements, limits should be placed on the allowable sound power emitted by a pump or motor as a function of horsepower.
4. Because of the large sample standard deviation of sound output from hydraulic components, allowable limits should be based on the mean of three measurements.
5. Further studies should be conducted to determine if a practical pump's noise rating can be used for selecting pumps so that the total amount of testing is minimized and the selection process simplified.

APPENDIX A

TEST PROCEDURES

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APPENDIX A

TEST PROCEDURES

The three parameters of interest regarding noise in fluid power systems are airborne noise, structureborne noise, and fluidborne noise. This appendix discusses the procedures used to determine the parameters of interest for the purposes of this study.

AIRBORNE NOISE

The sound power measurements contained in this report were obtained in the OSU-FPRC Reverberant Test Facility, whose characteristics have been discussed in previous reports [1][2][3]. The procedures used for the sound power determinations were discussed in the same references. It should be noted that the test procedure relied heavily on the National Fluid Power Test Procedures for determining the sound emitted by hydraulic pumps. The NFPA Procedures were tempered by the knowledge gained through the efforts of Working Group 1 to ISO Subcommittee 8, Technical Committee 131. Members of the FPRC staff were fortunate enough to participate with Mr. Paul Hopler and other representatives to Working Group 1 in the development of a new ISO Recommended Procedure for measuring the sound emitted by hydraulic pumps. Many of the ideas included in the new ISO procedure were used for the measurements reported in this study.

The measurements of the filter stand noise were made on a relative basis in a non-verified acoustical environment. All of the measurements were made at the same position, with the only principal change being those noted in the graph displaying the test results.

FLUIDBORNE NOISE

A fluidborne noise test procedure for pump pressure ripple was outlined in last year's report [3]. For each test, the load system for the pump was held constant, so that the measurements can be compared on a relative rather than an absolute basis. The pressure transducer was placed as close as practical to the outlet of the pump, with the load valve being approximately ten feet from the pump outlet. The test fluid was Mil-2104. Because the efforts of MERDC and other interested parties in the fluid power industry, the NIEP is now undertaking a study to develop a Recommended Test Procedure for determining fluidborne noise level of a fluid power pump. It is anticipated that this test procedure will provide the users of fluid power pumps with a practical means of discriminating between components in order to select "quieter" sources.¹

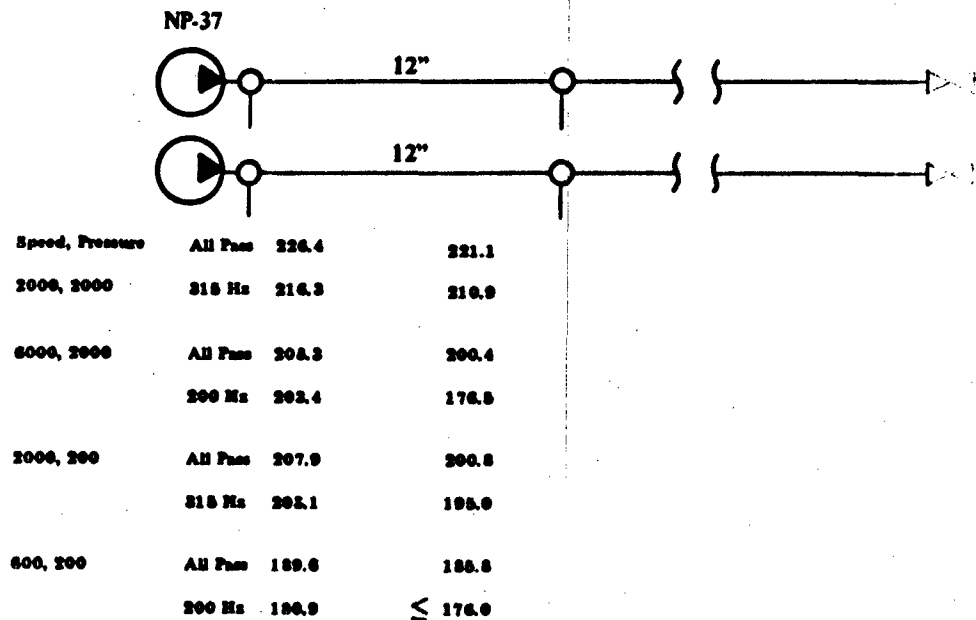
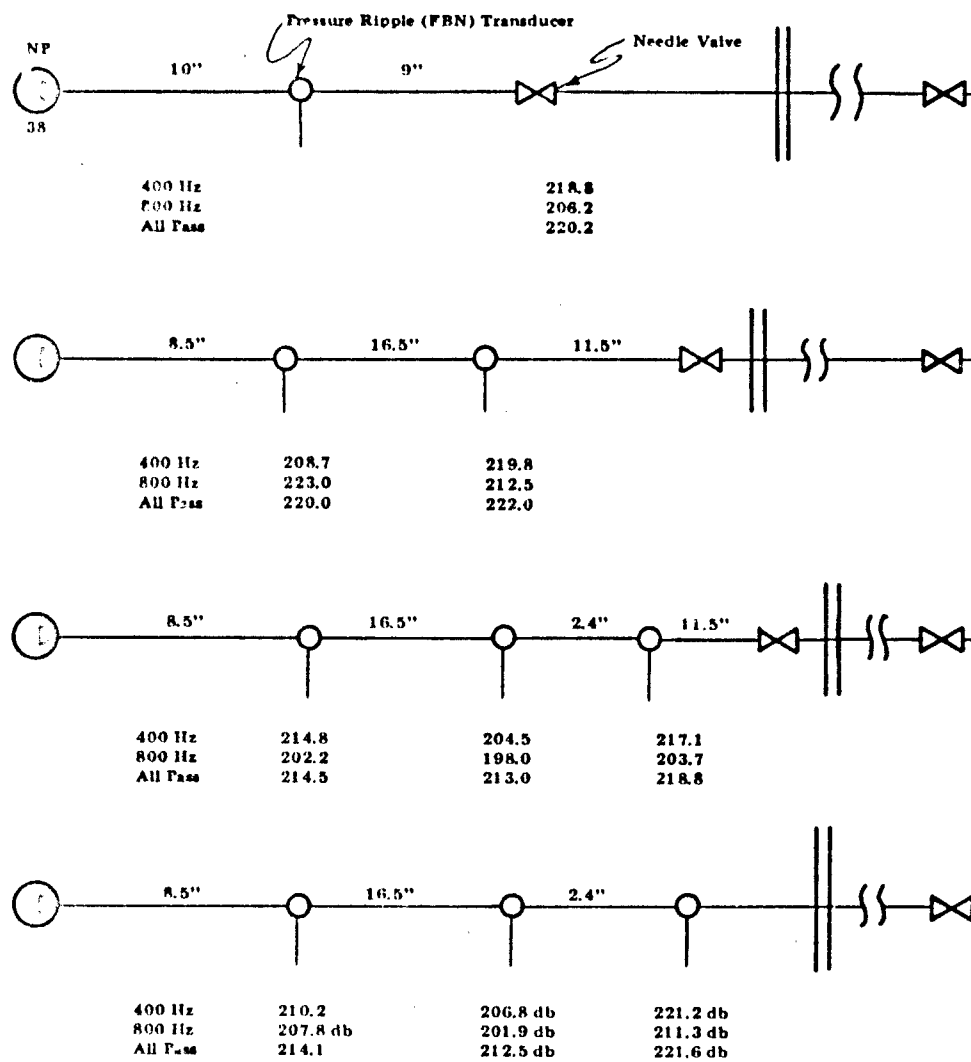


Fig. A-1. Comparison of FBN Levels Near Pump and Approximately One Foot From Pump. (db re 20μ/M²)

Fig. A-1 and A-2 show some pressure ripple measurements which (A-1) demonstrate the possibility that FBN measured at the pump are maximum and (A-2) show that "standing waves" do indeed exist in fluid conduits.



NP-38, 2000 rpm, 2000 psi, 65.5 °C, Mil-2104, 5 psi Inlet. Add 124 db to obtain db re 20 μN/M². Individual frequency analyzed with 1/3 octave filters. (740624)

Fig. A-2. FBN Level Changes Due to Load Valve Location.

The fluidborne noise measurement techniques used for the evaluation of the FBN attenuators is described in Ref. [7]. Basically, the procedure consisted of locating three transducers upstream and three transducers downstream of the component to be evaluated. The distance between the transducers was approximately five feet. This distance insured that a reasonably adequate sampling of the pressure ripple energy was possible. For transmission loss measurements, both the upstream and downstream transducers were monitored and each set of outputs was averaged to obtain effective upstream and downstream levels. For insertion loss measurements, only the downstream transducers were averaged to indicate the effective pressure ripple level with and without a particular attenuator.

VIBRATION MEASUREMENTS

Single-point measurements were used for all structureborne data. For each test, a vibration calibrator was used which set the level of 100 db relative to 1g at 100 Hz. All of the vibration measurements included in this report are intended to be considered on a relative basis; thus, no detailed test procedure was prepared to accompany the structureborne data.

APPENDIX B

ACOUSTICAL DATA REDUCTION

APPENDIX B

ACOUSTICAL DATA REDUCTION

The computer programs used in this study for reducing raw data to dbA sound power, vibration, and fluidborne noise levels were essentially the same as those reported last year [3]. Details of the programs are contained in that report.

One additional computer program was used to obtain the parameters for the pump sound power program. This program, called PUMPAR, is shown on the following pages of this appendix. PUMPAR uses a least-squares fit of the experimental data to yield the desired pump model coefficients. In its present form, the program works best with four data points. It is recommended that, for future development work, consideration be given to preparing a new program. The model parameters can be derived from as few as three data points. This can be reflected in a revised program.

```

10. //PUMPPARA JOB (12687,447-48-9533,1),'JOHN'
20. /*ROUTE PRINT RJO
30. // EXEC FORTGCG
40. //FORT.SYSIN DD *
50.     DIMENSION NP(100),RPM(100),OP(100),WP(100),INT(100),X(100),X1(100)
50.5     DIMENSION Y(100),Y1(100),Y2(100),A(100),PW(100),X2(100)
50.6     DIMENSION AA(100)
51.     WRITE(6,2)
52.     2 FORMAT(2X,'PUMP ID',2X,'R EQUALS',7X,'A EQUALS')
53.     WRITE(6,3)
54.     3 FORMAT(4X,'K EQUALS',7X,'MFAS. SP',7X,'CALC. SP')
60.     500 DO 200 K=1,2
70.         SY=0
80.         SX=0
90.         SXX=0
100.        SXY=0
110.        DO 100 I=1,2
120.            READ(5,1)NP(I),RPM(I),OP(I),WP(I),INT(I)
130.            X(I)=OP(I)
140.            Y(I)=ALOG10(WP(I))
150.            SX=SX+X(I)
160.            SY=SY+Y(I)
170.            SXX=SXX+X(I)**2
180.            SXY=SXY+X(I)*Y(I)
190.        100 CONTINUE
200.        A(K)=(2*SXY-SX*SY)/(2*SXX-SX**2)
210.        200 CONTINUE
220.        B=(A(1)+A(2))/2
230.        DO 300 J=1,2
240.            SX=0
250.            SY=0
260.            SXX=0
270.            SXY=0
280.            DO 700 M=1,2
290.                READ(5,1)NP(M),RPM(M),OP(M),WP(M)
300.                X1(M)=ALOG10(RPM(M))
310.                Y1(M)=ALOG10(WP(M))
320.                SX=SX+X1(M)
330.                SY=SY+Y1(M)
335.                SXX=SXX+X1(M)**2
340.                SXY=SXY+X1(M)*Y1(M)
345.            700 CONTINUE
350.            AA(J)=(2*SXY-SX*SY)/(2*SXX-SX**2)
355.        300 CONTINUE
360.        AAA=(AA(1)+AA(2))/2
370.        SY=0
380.        SX=0
390.        SXX=0
400.        SXY=0
410.        DO 400 L=1,4
420.            READ(5,1)NP(L),RPM(L),OP(L),WP(L),INT(L)
430.            X2(L)=(RPM(L)**AAA)/(10**(P*OP(L)))
440.            Y2(L)=WP(L)
450.            SXX=SXX+X2(L)**2
460.            SXY=SXY+X2(L)*Y2(L)
470.            SX=SX+X2(L)
480.            SY=SY+Y2(L)
490.        400 CONTINUE
500.        PK=(4*SXY-SX*SY)/(4*SXX-SX**2)
510.        1 FORMAT(12,2X,F5.0,2X,F5.0,2X,F7.0,2X,11)
520.        DO 600 N=1,4
530.            PW(N)=(PK)*(RPM(N)**AAA)/(10**(B*OP(N)))
540.            WRITE(6,11)NP(N),B,AAA,PK,WP(N),PW(N)
550.            11 FORMAT(4X,14,1PE15.3,F15.3,E15.3,E15.3,E15.3)
560.        600 CONTINUE
570.

```

580.				
590.				
600.				
650.	25	600.	200.	5.58E-6
660.	25	600.	2000.	1.51E-5
670.	25	2000.	200.	3.40E-5
680.	25	2000.	2000.	1.50E-4
690.	25	600.	200.	5.58E-6
700.	25	2000.	200.	3.40E-5
710.	25	600.	2000.	1.51E-5
720.	25	2000.	2000.	1.50E-4
730.	25	600.	200.	5.58E-6
740.	25	600.	2000.	1.51E-5
750.	25	2000.	200.	3.40E-5
760.	25	2000.	2000.	1.50E-4
940.	//			

APPENDIX C

SELECTED REFERENCES

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APPENDIX C

SELECTED REFERENCES

1. *"Hydraulic Specification Program (U),"* Phase II, Annual Report No. OSU-FP-111, Prepared for the U.S. Army Mobility Equipment Research and Development Center, Ft. Belvoir, Virginia, Contract No. DAAK02-71-C-0074, December 1971.
2. *"Hydraulic System Noise Study (U),"* Section III, Annual Report No. FPRC-2M3, Prepared for the U.S. Army Mobility Equipment Research and Development Center, Ft. Belvoir, Virginia, Contract Nos. DAAK02-71-0074 and DAAK02-72-C-0172, November 1972.
3. *"Hydraulic System Noise Study (U),"* Section III, Annual Report No. FPRC-3M3, Prepared for the U.S. Army Mobility Equipment Research and Development Center, Ft. Belvoir, Virginia, Contract No. DAAK02-73-C-0172, December 1973.
4. Maroney, G. E. and L. R. Elliott, *"An Acoustical Performance Appraisal Technique for Fluid Power Pumps,"* SAE Earthmoving Industry Conference, Peoria, Illinois, April 1974.
5. Maroney, G. E., *"Acoustical Signature Analysis As a Non-Intrusive Diagnostic Tool,"* Basic Fluid Power Research Program, Annual Report No. 8, Paper No. P74-4, Fluid Power Research Center, Oklahoma State University, October 1974.
6. *"Noise Suppression Study of a Military D7F Tractor,"* Final Report for the U.S. Army Mobility Equipment Research and Development Center, Ft. Belvoir, Virginia, Contract No. DAAK02-72-C-0014, Caterpillar Tractor Co., Peoria, Illinois, September 1972.

7. *"An Investigation of the Fluidborne Noise Attenuation Characteristics of the 15 Cubic Inch Pulse-Tone,"* Prepared for Greer Hydraulics, Inc. by the Staff of the Fluid Power Research Center, Oklahoma State University, Report No. FPRC 74-4G-2.
8. *"A Study of the Performance Characteristics of An Acoustic Attenuator for Fluid Power Systems,"* Prepared for J. I. Case Company by the Staff of the Fluid Power Research Center, Oklahoma State University, Report No. FPRC 72-2, June 1972.
9. Bulletin 206-F, PULSCO Division, American Air Filter Company, Santa Paula, California, February 1972.

APPENDIX D

INSTRUMENTATION

APPENDIX D

INSTRUMENTATION

I. GENERAL RADIO

A.	1523	Level Recorder
B.	1523-P1	Preamplifier Plug In
C.	1523-P3	1/3 Octave Band Analyzer
D.	1523-P4	Wave Analyzer
E.	1523-9621	25 db Potentiometer
	1523-9622	50 db Potentiometer
	1523-9624	100 db Potentiometer
F.	1560-9531	Microphone
G.	1560-9580	Tripod
H.	1560-9666	Microphone Cable
I.	1560-P13	Vibration Pickup System
J.	1560-P42	Microphone Preamplifier
K.	1562-A	Sound Level Calibrator
L.	1382	Random Noise Generator
M.	1933	Precision Sound Level Meter and Octave Band Analyzer
N.	135A	X-Y Recorder
O.	130BR	Oscilloscope
P.	400 LR	Vacuum Tube Voltmeter
Q.	1557-A	Vibration Calibrator.

II. HEWLETT-PACKARD

A.	3300-A	Function Generator
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II. HEWLETT-PACKARD (Cont.)

- B. 205AG Signal Generator
- C. 202CR Signal Generator
- D. 410B Vacuum Tube Voltmeter

III. BOGEN

- A. CH13-35A Amplifier

IV. TEKTRONIX

- A. 502 Dual-Beam Oscilloscope
- B. RM31A Oscilloscope

V. PCB PIEZOTRONICS, INC.

- A. 118A02 Quartz Pressure Transducer
 - 1. SN 646 0.224 pcb/psi 11.5 pfd Capacitance
 - 2. SN 647 0.224 pcb/psi 11.5 pfd Capacitance
 - 3. SN 648 0.224 pcb/psi 11.5 pfd Capacitance
 - 4. SN 649 0.260 pcb/psi 11.5 pfd Capacitance
 - 5. SN 650 0.254 pcb/psi 11.0 pfd Capacitance
 - 6. SN 651 0.242 pcb/psi 11.5 pfd Capacitance
 - 7. SN 297 0.248 pcb/psi 18.0 pfd Capacitance
- B. 402A Pressure Amplifier
 - 1. SN 1120 0.992 gain 4.5 pfd Capacitance
 - 2. SN 1118 0.993 gain 4.5 pfd Capacitance
 - 3. SN 1110 0.992 gain 4.5 pfd Capacitance
 - 4. SN 1111 0.993 gain 4.5 pfd Capacitance
 - 5. SN 1109 0.993 gain 4.5 pfd Capacitance
 - 6. SN 1119 0.985 gain 4.5 pfd Capacitance

V. PCB PIEZOTRONICS, INC. (Cont.)

- B. 402A Pressure Amplifier
 - 7. SN 815 0.994 gain 204.5 pfd Capacitance
 - 8. SN 563 0.992 gain 204.5 pfd Capacitance
- C. 482-A ICP Power Supply
- D. 483A02 ICP Power Supply

VI. BELL & HOWELL

- A. 4-402-0001 Pressure Transducer

VII. DAYTRONIC

- A. Type 91 Strain Gauge Transducer
Input Module
- B. Model 300 Transducer Amplifier Indicator
- C. Type P Galvanometer Driver Output
Module

VIII. KENWOOD

- A. KA-4004 Amplifier

IX. TEAC

- A. 1230 Tape Deck

X. BECKMAN

- A. 7370R Universal EPUT Meter

XI. KROHN-HITE

A. 440AR Oscillator

XII. RUTHERFORD

A. B7B Pulse Generator

APPENDIX E

MATERIALS AND COMPONENTS

APPENDIX E

MATERIALS AND COMPONENTS

This appendix contains a list of the various components tested for this study. Also included in this appendix are the various acoustical materials used for noise control purposes:

1. Pipe and Valve Covering Product Number 45112, Sound Solutions TM Corporation.
2. PULSCO FBN Attenuator, American Air Filter, APT-4.
3. Pulse-Tone FBN Attenuator, Greer Hydraulics, 15 Cubic Inch.
4. V101S4S11D20L Hydraulic Pump, U.S. Army MERDC.
5. V2F1F9S18B8H20A001L Hydraulic Pump, U.S. Army MERDC.
6. Sound-Off, Quaker State, Damping Compound.
7. Good Vibrations TM High Temperature Damping Compound, 70306, Sound Solutions TM.
8. Damping Compound, Kustics, KDC-E-162.
9. V2520 Hydraulic Pump, U. S. Army MERDC.
10. "Duct Board" Rigid Fiberglass with Aluminum Back, Owens-Corning Type 475-FR9SD
11. Leaded Vinyl, John Schuller and Associates, Sound/Ease tlb-M, tlb-1.
12. Leaded Vinyl, Singer Partitions, Inc., Super Sound Stopper.
13. Foam Rubber, 2 in. Thick, 21 oz./ft.³ (21,000 gm/m³)
14. Vibration Isolator Barry Number C-2150.

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